CHARACTERISTIC OF DRIFT ELIMINATORS OF AN EVAPORATIVE CONDENSER WITH FORCED AND INDUCED FLOW

by

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NOVEMBER, 1989

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A Thesis Submitted
in Partial Fulfilment of the Requirements
for the Degree of

MASTER OF TECHNOLOGY

by

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to the

DEPARTMENT OF MECHANICAL ENGINEERING
INDIAN INSTITUTE OF TECHNOLOGY KANPUR

NOVEMBER, 1989

TO MY

PARENTS, BROTHERS AND SISTERS

CERTIFICATE

This is to certify that the work entitled 'CHARACTERISTICS OF DRIFT ELIMINATORS OF AN EVAPORATIVE CONDENSER WITH FORCED AND INDUCED FLOW' has been carried out by Mr. Amit Kumar Singh under my supervision and has not been submitted elsewhere for the award of a degree.

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Nov. 30, 1989

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ACKNOWLEDGEMENTS

I wish to express my deep sense of gratitude to Dr. Keshav Kant for giving me this opportunity to work under his valuable guidance, for his keen interest and encouragement throughout the course of this project.

I am also thankful to Dr. Manohar Prasad for his moral support and encouragement during my stay over here.

I also take this opportunity to thank Mr. P.N. Mishra and Mr. Sushil Kumar Mishra for providing me all kinds of assistance in seting up the Test Rig.

I am also thankful to my friend Alak Kumar Betal for providing necessary help for conducting the experiment.

I also wish to thank Mr. R.C. Vishwakarma for his excellent typing as well as his help in the fabrication of Test Rig.

AMIT KUMAR SINGH

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NOMENCLATURE

C_1	Cost of drift loss, paise per hour
C_2	Cost of power loss, paise per hour
CDE	Concrete drift eliminators
COP	Coefficient of performance
g	Acceleration due to gravity, ms^{-2}
h	Pressure drop in mm of water
$\dot{m}_a$	Mass flow rate of air $kg.min^{-1}$
m_d	Specific drift loss, kgw/kgda
M_d	Rate of drift loss, kg h^{-1}
m_e	Specific evaporation loss, kgw/kgda
$\dot{m}_w$	Water mass flow rate, kg min^{-1}
n	Number of stages
p	Static pressure at a point, mm of water
p_d	Discharge pressure of compressor, bar
p_s	Suction pressure of compressor, bar

P_{com}	Power input to the compressor, W
P_{FD}	Power input to FD fan, kW
P_{ID}	Power input to ID fan, kW
t	Dry Bulb Temperature (DBT), ^{o}C
t^*	Wet Bulb Temperature (WBT), ^{o}C
T_1	Condenser inlet temperature, ^{o}C
T_2	Condenser outlet temperature, ^{o}C
V_{vol}	Volumetric discharge of air, $m^3\ h^{-1}$
V	Supply voltage, V
w	Specific Humidity, kgw/kgda
w_1	Ambient air specific humidity, kgw/kgda
w_2	Specific Humidity of the leaving air without duct heater, kgw/kgda
w_3	Specific Humidity of the leaving air with duct heater, kgw/kgda
WDE	Wooden drift eliminators
ρ_w	Density of water, kg/m^3
Δp	Total pressure drop, mm of H_2O
θ	Inclination Angle, degree

ABSTRACT

A systematic experimental study was taken up on Drift eliminators used in Evaporative Condensers to determine the pressure drop across them, drift loss and its effect on system performance. The wooden and concrete drift eliminator plates were made and mounted in a frame which comprised a stage of the drift eliminators. Several stages like this were made for the current study. The experiments were conducted first with a single stage wooden drift eliminators (WDE) and then with double stages. The orientation of drift eliminator plates was varied from 15° to 90° for one single fan speed. To alter the fan speed, the supply voltage was varied from 160V to 230V AC. Pressure drop and drift loss data were collected with FD and ID fans for various orientations of drift eliminator plates and for a range of fan speed. The above sets of experiments were repeated for single stage and double stages of concrete drift eliminators (CDE) and all the required data were collected.

Based upon the cost analysis, the variation of the pressure drop and the drift loss suggested an optimum angle of inclination of the plates. The optimum value depends upon

the number of stages used at a time. As the number of stages goes up, the optimum value of θ moves towards higher side.

The COP of the system increased with increase in fan speed and angle of inclination and decreased with increase in the number of stages. The use of FD fan resulted in better performance than that of ID fan.

CHAPTER 1

INTRODUCTION

1.1 GENERAL BACKGROUND

In evaporative cooling, water is sprayed into a flowing air media and the temperature of the water is reduced because of the evaporation of water and subsequent absorption of latent heat by the air stream. Typical industrial applications where this principle is used are the cooling towers in power plant or central air conditioning systems, evaporative coolers for process industries and desert coolers. In case of evaporative condensers, the water which is sprayed over the condenser tubes takes away some heat and secondly the air whose temperature is slightly increased due to the evaporation of water droplets, also removes some heat from the condenser tubes. The heated water past the condenser is cooled by the process of adiabatic saturation.

In recent years, the demand for industrial water has increased and at the same time, the sources of raw water are limited. To tackle this problem, we need an efficient mechanical system to retain the water since the spray or the water distribution system turns the process water into light droplets before they are cooled by the air stream. As the air moves counter to

or cross-wise to the flow of water, it will pick up much of the mist and droplets and carry them with the air flow out of the tower or the evaporative condenser. This is known as drift loss. One way of tackling this problem of drift loss is by providing the drift eliminators above the condenser coil in an evaporative condenser and above the spray system in a cooling tower. The drift eliminators are typically rectangular plates which are set at the desired angle to the air flow direction over the whole outlet area of the air flow passage. By adjusting the angles of these plates in the flow direction suitably, most of the water droplets can be made to fall back into the sump, thereby reducing the drift loss.

Cooling tower performance, i.e., heat removal, is a balance between water flow and air volume, therefore drift eliminators are normally designed to be efficient through a calculated range of air flow. Too great an air speed can result in excessive drift loss of water from the tower, while poorly designed eliminators will adversely affect the performance of the unit. Thus drift eliminator effectiveness is an essential aspect of cooling tower design for many reasons, among than a few are:

1. conservation of water;

2. retention of chemicals used for the treatment in the water sump.

3. prevention of staining by chemical additives eg. chromates etc.

4. avoiding fan blades corrosion in case of induced draft.

5. avoidance of violation of local area environmental protection regulations.

The present study concerns with the performance characteristics of multistage wooden as well as concrete drift eliminators with FD and ID fans separately; for an evaporative condenser and its effect on the performance of a refrigeration system.

1.2 LITERATURE SURVEY:

The principles of evaporative condenser has been known for a long time. Because of the wide use that such a condenser was put to in industrial applications, most books on Refrigeration and Airconditioning mention about evaporative condenser (Stoecker, 1982; Prasad, 1985). Another common equipment which makes use of the principle of evaporative cooling is the cooling tower.

Evaporative condenser may be preferred over cooling towers for small capacities but for larger capacities there is no other alternative but to use a cooling tower if conservation of water is very important. In the former as mentioned earlier the condensation takes place because of the heat extraction by the cold water stream falling over the tubes as well as by the relatively cold air stream. In the latter though, only the cooling of the water takes place inside the cooling tower by the process of adiabatic saturation, but the actual heat transfer from the condenser takes place away from the tower through indirect contact (Perry, 1973;Goodman, 1938).

Evaporative cooling with water recycling is a viable method of making efficient use of cooling water. Another method which finds use in small scale application is purely the cooling of condenser by air. The lowest condenser temperature, achievable in this fashion is equal to the Dry Bulb Temperature (DBT) of the entering air. But,by using the evaporative cooling process, the lowest condenser temperature can be brought close to the Wet Bulb Temperature (WBT) of the entering air. Because the wet bulb temperature is always less than or equal to dry bulb temperature, the refrigeration system can be operated with lower condensing temperature thereby leading to a higher COP, of the cycle.This can enhance the

cycle efficiency considerably particularly, in a non-humid hot climate, i.e., in deserts (Perry, 1973; Threlkeld, 1962).

In evaporative cooling, the heat-transfer process involves (1) latent heat transfer owing to vapourisation of a small portion of the water and (2) sensible heat transfer owing to the difference in temperature of water and air. Approximately 80% of this heat is due to latent heat and 20 percent to sensible heat. Theoretical possible heat removal per unit quantity of air circulated through an evaporative cooling equipment depends on the temperature and moisture content of air. An indication of the moisture content of the air is its wet-bulb temperature which is the lowest theoretical temperature to which the water can be cooled. Practically, the cold-water temperature approaches it, but does not equal the air wet-bulb temperature in the evaporative cooling process; this is because it is not possible to contact all the water with fresh air as the water drops through the evaporative condenser. So the important factors for an efficient evaporative cooling process are air to water contact (residence) time, and breakup of water into droplets (Boelter, 1939).

One of the problems associated with the evaporative cooling process is the carry over of water droplets with the air flow which is formed as 'drift loss'. This drift loss is

undesirable as it is a direct loss of water to the atmosphere. Particularly in places where there is a tremendous scarcity of water, this may result in a very big expense. The drift loss is also undesirable because of setting of water droplets on any electrical equipment may prove to be hazardous. Also in the case of Induced Draft Fan the water particles may intensify the corrosion of the blades thereby reducing the life of the fan. For these reasons, the thermal pollution created by the transported water droplets should be reduced.

Studies have been made on cellular drift eliminators for cooling towers. The material used for construction of cellular drift eliminator is neoprene-asbestos. A cellular drift eliminator installtion adds to the efficiency of the operation by stopping the air from making heavy turns arround herringbone, and also collects any droplets which are carried away by outgoing air. An additional bonus of neoprene-asbestos material is that it provides a fire proof barrier. Thus, a match, lighted cigrette, or burning rag from an incinerator that may land on top of the drift eliminators when tower is not in operation will not start a conflagration because the fire will extinguish itself on the asbestos (Burger, 1975).

Although evaporative condensers have been in use for a long time, till to date systematic studies have not been reported on the efficient use of drift eliminators in them. Details of the reduction in Drift Loss and the associated increase in power consumption while drift eliminators are used, have not been analysed. The present work attempts to investigate some of these aspects of the drift eliminator characteristics.

1.3 SCOPE OF THE PRESENT WORK:

The objectives of this research project are as follows

1. Determination of the 'Pressure Drop' and 'Drift Loss' for one and two stages of the 'Drift Eliminators' for various orientations of the drift eliminator plates made of wood and concrete, in an evaporative condenser.

2. Studying the variation of ' Pressure Drop' across the drift eliminators and'Drift Loss' for various air flow rates through the evaporative condenser.

3. Determination of an optimum angle of orientation for drift eliminators plates for the given capacity of the condenser and given geometry of the plates.

4. Studying the effect of drift eliminators on the performance of a refrigeration system.

5. Comparative study on the use of ID and FD fans in an evaporative cooling system.

For the data in this study, the refrigeration system capacity, water circulation rate and the type of spray system were kept unchanged.

1.4 ORGANIZATION OF THE THESIS:

Chapter two describes the experimental methodology including the details of instrumentation and the medhod of estimating the drift loss. Chapter three describes the drift loss, pressure drop and refrigeration system performance data collected during the course of this investigation. Discussion of the results including the main conclusions and suggestions for future resrarch are also given.

CHAPTER 2

EXPERIMENTAL METHODOLOGY

2.1 TEST RIG:

The complete test rig consists of an evaporative condenser using a forced draft as well as an induced draft fan. The induced draft fan is mounted on a raised platform on the top of the test rig. The drift eliminators used are of two kinds; namely wooden and those made of concrete. The evaporative condenser has been installed as one of the components of a R-22 refrigeration system. The main components of the Test Rig are shown in Figure 2.1. Their description and the specifications are given below:

1. REFRIGERATION SYSTEM: Consists of the following

   a. Compressor:

   It's a SHRIRAM 1.5 ton capacity R-22 compressor.

Specification	:	1622
Supply Voltage = 220V AC, Amps	=	12.2, RPM = 2850,
Evaporating Temperature	=	7.2^{o}C,
Condensing Temperature	=	55^{o}C

Ambient Temperature	= $35^{o}C$,
Compressor Suction Gas Temperature	= $35^{o}C$,
Suction Pressure	= 5.25 bar (75.95 PSIG)
Discharge Pressure	=21.467 bar or (300.54 PSIG),

b. Condenser:

Designed for a capacity of 1.5 ton. It is made of 3/8" (9mm) copper tube of length 23.10 m.

c. Capillary Tube:

It was provided for a rated capacity of 1.5 ton as an expansion device. The capillary tube length in 1.50 m.

d. Evaporator (Cooling Coil):

The evaporator is 1.5 ton finned unit. A fan was installed at the back of this unit to blow ambient air for better heat transfer to the coil.

2. MAIN CHAMBER

Main chamber consists of (1) a rectangular box (1.00m x 0.90 x 1.40 m) (2) a drift eliminator chamber (1.00 m x 0.52m x 0.52 m) having top portion taperred and (3) a heater box (0.45m x 0.45 m x 0.45m). The main chamber is made of angle iron frame and aluminium sheet. The Drift Eliminator chamber houses drift eliminators supported on angles. The heater box houses 6 kW capacity finned electric duct heater on top of drift eliminator chamber. The bottom portion of the main chamber was used as a water sump as shown in Figure 2.1.

3. DUCTS:

A main duct was connected at the top of the heater box to the inlet of ID fan mounted on a raised plat-form by the side of main chamber. The discharge duct was connected at the out let of ID fan to carry the discharge air out of the room. The dimensions of discharge duct are 0.410m x 0.320m x 2.26m and was equipped with a damper to control the flow in the sample duct. Sample duct is of dimensions 0.18m x 0.18m x 1.20 m which is used to measure the psycorometric condition of air at its outlet. The main duct has two parts

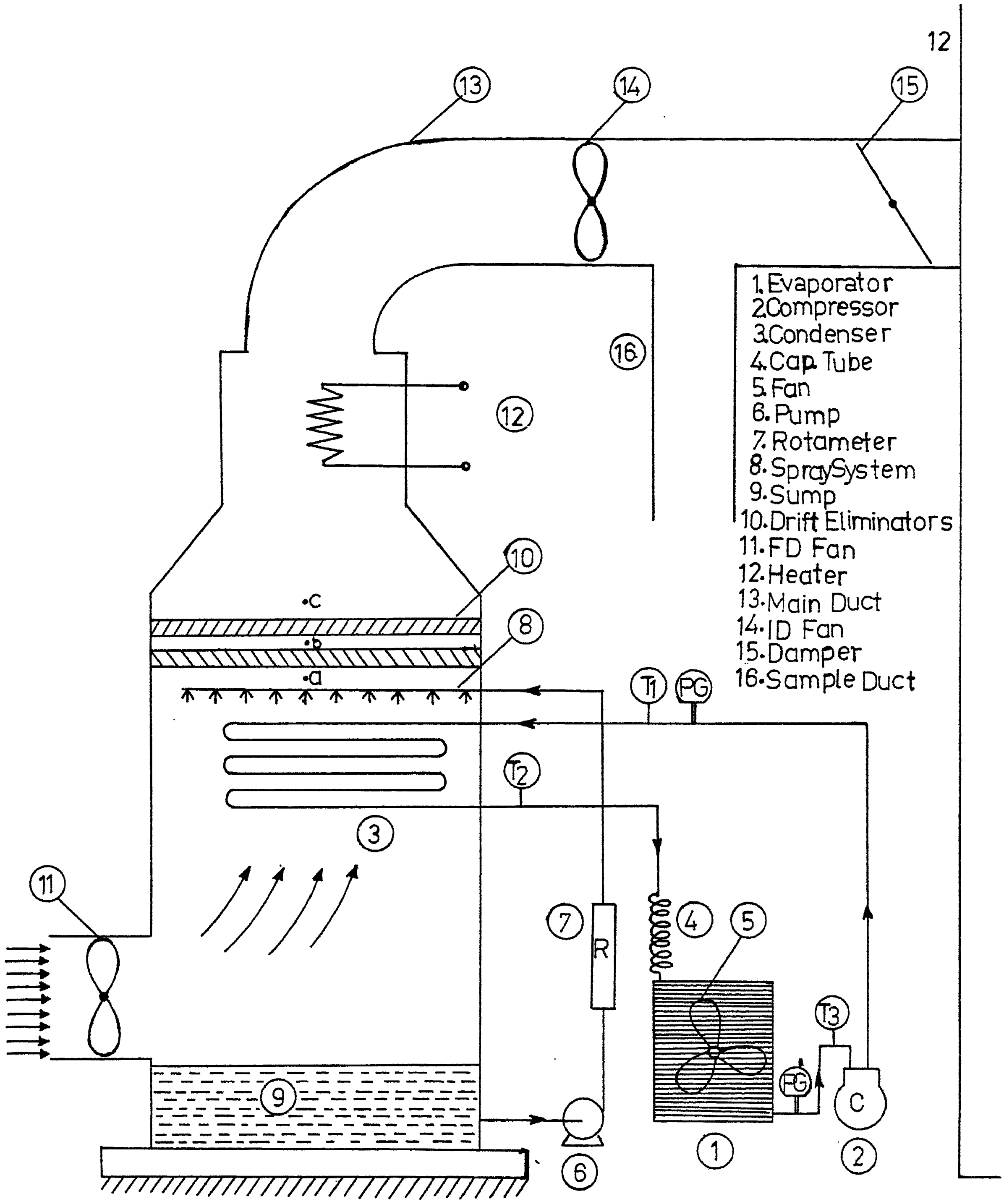

Fig. 2.1
Schematic Diagram of the Test Rig

(1) a 90^{o} duct of cross section area 0.46m x 0.28m and (2) a connecting duct of rectangular cross-section of 0.46m x 0.28m at one end and a circular cross-section of dia 36 cm at the other end. The duct is 1.30 m long.

4. DRIFT ELIMINATORS:

These were provided to reduce the drift loss. In the present case we have used two types of drift eliminators (1) Wooden drift eliminators (WDE) and (2) concrete drift eliminators (CDE). In case of wooden drift eliminators each strip can be rotated from 0^{o} to 180^{o}. For concrete drift eliminators a separate rotating device was used to change the orientation.

(a) Wooden Drift Eliminators:

Specification : Area of box 0.495m x 0.95 m

Thickness of each strip	= 13 mm
Number of strips in single stage	= 9
Width of the each strip	= 46 mm

(b) Concrete Drift Eliminators:

Thickness of each drift eliminator	= 25 mm
Length of each drift eliminator	= 900 mm
Width of each drift eliminator	= 50 mm
Number of drift eliminators in stage	= 7

5. CIRCULATING WATER PUMP:

It was installed to provide the spray water over the condenser coil through a sprayer consisting of 4 rows of 1/2" conduite pipe with holes of size 2 mm dia. The pump specifications are as follows

HP = 2 , 3 ∅, Volt. 230/240

RPM = 2840 , Amp = 2.5

Make = Alfa Laval

Type = EM 1D 111B

6. I.D. FAN AND F.D. FAN:

F.D. fan and motor assembly was installed on a concrete foundation specially made for it by 6 foundation bolts.

Specifications: Fan discharge $89.9 m^3/min$, static pressure 13.5 mm H_2O 250 mm diameter, Motor o.75 hp., 1420 rpm, 5.2 amp. 230 Volt., single phase. (Premier Corporation India Ltd. Coimbotore).

I.D. fan specifications are same that of F.D. fan.I.D.fan is mounted on a raised platfrom supported on two 'L' shape grounded 4" C.S.pipes.

2.2 INSTRUMENTATION USED:

Instruments were used to measure pressure and temperatures at different points in the refrigeration system. The pressure drop across the drift eliminators was measured by using a manometer directly calibrated in inches of water column. The specifications of various gauges used are given below.

a. Compressor Suction Side:

Pressure Gauge Range	0 - 180 PSIG
Temperature	Copper-constanton themocouple.

b. Compressor Discharge Side:

Pressure Gauge Range	0 - 300 PSIG
Temperature	Copper-Constanton themocouple

c. Condenser Outlet Side:

Temperature	Copper-Constanton themocouple

d. A vacuum pump was used to evacuate the refrigeration system before charging the refrigerant.

e. In order to determine the rate of water spray over the condenser coil, a Rotameter (0-5 gpm, specific gravity 1.0) was used to measure the circulating water flow rate. For a check a water tank of known volume capacity and a stop watch for measuring the time were also used to measure the flow rate of sprayed water on condenser.

f. The psychrometric data of the air entering and leaving the evaporative condenser were measured using an ordinary psychometer (DBT -20 to 50°C, WBT -20°C + 50°C). To be able to measure these data for the leaving air, the damper of the main duct was closed partially so that the sufficient amount of air could flow through the sample duct. One Hygro-Thermograph (M 594 Mfd The Bendix Corporation, Maryland U.S.A.) was also used to measure DBT and Relative Humidity directly.

g. Velocity of air was measured using a Vane Anemometer (0 to 1 x 10^5m, wind speed 1 to 15 m/s, OTA KEIKI SEISAKUSHO, JAPAN) at the inlet duct to the F.D. fan.

h. The power input to the compressor, FD and ID fans was measured by using Wattmeters.

Wattmeter (Compressor)-	Make	: Nippen
	Class	: (1.5) I.S. 1248-68
	Model	: SF-144P-1EW
	Amp.	: 20 A
	Voltage	: 250 V
	Range	: 0- 1500 W
Wattmeter (FD and ID fan)	Make	: Toshniwal
	Range	: 0-2.5 kW
	Least count	: 0.05 kW

i. The supply voltage to the FD and ID fans was varied from 160 V to 230V bt using a Dimmerstat (0-270 Volts, max. load 15 amps, Mfd Automatic Electric Private Ltd- Bombay).

j. The pressure drop across the drift eliminators was measured using a Manometer (Range - 0.1" to +1.0",Least count-0.02", Make-Dwyer, Sp. gravity of oil - 0.826).

k. The temperatures sensed by thermocouples were read form an electronic temperature recorder (Honeywell Temperature Recorder Electronik-15, Range-0-300°F).

2.3 EXPERIMENTAL PROCEDURE:

Various steps of the experimental procedure are outlined below:

1. Before starting the experimental run, all the electrical connections, water connections were checked to ensure safety of the setup.

2. Wooden drift eliminators which were set 90° to the horizontal were solid into the drift eliminator box.

3. The FD fan was run at 230V AC.

4. The pump was started for water supply to the spray system.

5. The refrigeration cycle was switched on and the pressure and temperatures at different points in the cycle were recorded. They were checked for not crossing the upper set limits.

6. After 15 minutes or so when the system was stablised, the suction and discharge pressures of the compressor, various temperatures of the refrigeration cycle, pressure drop across the drift eliminators using manometer, power consumed by the FD fan and compressor was recorded. The DBT and WBT were recorded for entering air at inlet to the main chamber and at the outlet of the sample duct.

7. The heater (6 kW) mounted on top of the drift eliminator box was switched on for evaporation of water droplets carried away with the out going air.

8. After about 15 minutes the readings were repeated for DBT and WBT at the out let of sample duct. Readings were also recorded by Hydrograph as a check. After the readings were taken heater was put off untill the next set of readings.

9. Above set of readings were repeated for inclination angles of 60°, 45°, 30° and 15°.

10. Similarly, steps 2 to 8 were repeated for the two stages of wooden drift eliminators with FD fan.

11. Then the ID fan was connected to the system and the experiment was repeated first.with single stage and then with double stages of wooden drift eliminators.

12. All the above data were repeated for concrete drift eliminators with FD and ID fans (of course one at a time).

13. Amount of water sprayed and also make up water were measured.

14. Measure the barometric pressure and ambient temperature.

The damper of the discharge duct was kept partially closed throughout the experimental runs .

2.4 ESTIMATION OF DRIFT LOSS:

Psychrometric data were measured for the entering air and air leaving through the sample duct without and with heater on. The damper of the discharge duct was kept partially closed.

1. MEASUREMENT OF EVAPORATION LOSS:

This is the amount of water loss during the cooling process. A simple mass balance of dry air and water over the evaporative condenser is given by

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a \text{ (say)}$$

$$\dot{m}_{a_1} w_1 + \dot{m}_e + \dot{m}_d = \dot{m}_{a_2} w_2 + \dot{m}_d$$

Where, $\dot{m}_{a_1}$, $\dot{m}_{a_2}$ = mass of dry air entering and leaving the evaporative condenser

t_1, t_2 = dry bulb temperature (DBT) of the entering and leaving air.

$\dot{m}_e$ = rate of evaporation of water (i.e. evaporation loss)

$\dot{m}_d$ = drift loss

From equations (2.4.1) and (2.4.2) we get

$$\dot{m}_e = \dot{m}_a (w_2 - w_1)$$

w_1 and w_2 can be easily determined knowing the DBT and WBT of the entering and leaving air. The mass flow rate of the dry air can be calculated knowing the discharge rating of the fan (Das, 1988).

2. MEASUREMENT OF THE DRIFT LOSS:

In order to measure m_d, the heater was switched on so that all the drift coming past the drift eliminators could be evaporated. This will change the quality of out going air. A simple mass balance over the evaporative condenser yields.

$$\dot{m}_a w_1 + \dot{m}_e + \dot{m}_d = \dot{m}_a w_3 \qquad (2.4)$$

or

$$\dot{m}_e + \dot{m}_d = \dot{m}_a (w_3 - w_1) \qquad (2.5)$$

Substituting from equation (2.4.3) into equation (2.5) yields

$$\dot{m}_d = \dot{m}_a (w_3 - w_2) \qquad (2.6)$$

and
$$\frac{\dot{m}_d}{\dot{m}_e} = \frac{w_3 - w_2}{w_2 - w_1} \qquad (2.7)$$

$\dot{m}_e$ and $\dot{m}_d$ can be easily determined by measuring the psychrometric data of the entering and the leaving air streams (Das, 1988).

CHAPTER 3

RESULTS AND DISCUSSION

In order to determine the drift loss from a varying number of drift eliminator stages and the pressure drop across them, experiments were conducted according to the precedure outlined in section 2.3.

3.1 DRIFT LOSS:

The angle of inclination (θ) was varied from 15° to 90° and for any given set of data, the drift eliminators were set at a particular angle. The number of stages used were two or one at a time. The dry bulb and wet bulb temperatures were measured for the air entering and leaving the evaporative condenser without and with duct heater. The psychrometric data of the moist air recorded during the experiments are given in Tables 3.1, 3.2, 3.3 and 3.4. Table 3.1 shows data for FD fan using single stage and double stages concrete drift eliminators. Table 3.2 shows data for ID fan using single stage and double stage concrete drift eliminators. Table 3.3 shows data for FD fan using single stage and double stage wooden drift eliminators while Table 3.4 shows data for ID fan

using single and double stage wooden drift eliminators. The evaporative (m_e) and drift (m_d) losses are computed and shown in Tables 3.5, 3.6, 3.7 and 3.8. It can be seen from these tables that the sp. drift loss increases if n decreases or θ increases. This is basically due to the fact that in either of the two cases (i.e. decreasing n or increasing θ), the net static pressure available for the flow increases which results in a higher volumetric discharge. The greater discharge of air brings larger amount of air indirect contact with water resulting in a larger value of m_e.

The specific drift loss, m_d is plotted versus θ for various supply voltages i.e., 230V, 200V, 160V. This is shown in Figures 3.1 through 3.6 covering single and double stages of both concrete and wooden drift eliminators for FD fan as well as ID fans. The trend of these curves is similar. As θ increases, the drift loss (m_d) increases, but it decreases with increasing number of stages (i.e. double stage). The drift loss as expected reaches a maximum for $\theta = 90^o$. As the fan RPM increases (with increase in supply voltage), the drift loss also goes up as shown in the figures mentioned above. It is seen that drift loss is more with FD fan than with ID fan of same capacity (see Figures 3.1 and 3.5). It is also observed from the data that the drift loss while using concrete drift eliminators is 20-25% less than that obtained using wooden drift

Table 3.1

Psychrometric Data of Moist Air Entering and Leaving the Evaporative Condenser for FD fan with CDE

Supply Voltage	Inclination Angle	No. of stages	Entering Air			Discharge Air without Heater			Discharge Air with Heater on		
			DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio
V	θ	n	t_1	t_1^*	W_1	t_2	t_2^*	W_2	t_3	t_3^*	W_3
(Volts)	(degree)		(^oC)	(^oC)	(kgw/kgda)	(^oC)	(^oC)	(kgw/kgda)	(^oC)	(^oC)	(kgw/kgda)
230	15	1	29.8	26.9	$21.0x10^{-3}$	30.6	28.7	$24.4x10^{-3}$	34.1	29.7	$24.8x10^{-3}$
		2	32.8	27.5	$21.3x10^{-3}$	32.7	29.0	$25.5x10^{-3}$	36.1	30.3	$25.8x10^{-3}$
	45	1	30.0	26.9	$21.0x10^{-3}$	31.1	29.1	$24.8x10^{-3}$	34.4	29.8	$25.8x10^{-3}$
		2	30.4	26.3	$19.8x10^{-3}$	30.4	28.3	$24.3x10^{-3}$	34.4	29.8	$25.4x10^{-3}$
	60	1	30.0	27.1	$21.6x10^{-3}$	31.0	29.2	$25.0x10^{-3}$	34.4	29.8	$25.8x10^{-3}$
		2	30.7	26.6	$20.4x10^{-3}$	30.9	28.6	$24.6x10^{-3}$	34.7	29.7	$25.6x10^{-3}$
	90	1	29.8	26.8	$21.0x10^{-3}$	30.5	28.7	$24.4x10^{-3}$	34.3	30.7	$26.6x10^{-3}$
		2	31.0	26.9	$20.6x10^{-3}$	31.3	28.8	$24.1x10^{-3}$	34.8	30.7	$26.6x10^{-3}$
	15	1	32.1	26.9	$20.2x10^{-3}$	31.4	28.9	$24.2x10^{-3}$	34.8	29.6	$24.5x10^{-3}$
		2	33.3	27.6	$21.2x10^{-3}$	32.2	29.5	$25.0x10^{-3}$	35.7	30.3	$25.4x10^{-3}$

200	45	1	31.8	26.5	$19.6x10^{-3}$	31.2	28.4	$23.6x10^{-3}$	34.9	29.3	$23.9x10^{-3}$
		2	33.5	27.7	$21.2x10^{-3}$	32.2	29.4	$24.4x10^{-3}$	35.8	30.3	$25.0x10^{-3}$
	60	1	31.4	26.7	$20.2x10^{-3}$	30.7	28.5	$23.6x10^{-3}$	34.6	29.3	$24.8x10^{-3}$
		2	33.4	27.8	$21.4x10^{-3}$	32.3	28.6	$24.4x10^{-3}$	35.9	30.3	$25.3x10^{-3}$
	90	1	30.6	26.6	$20.4x10^{-3}$	30.2	28.5	$24.0x10^{-3}$	31.8	28.7	$26.0x10^{-3}$
		2	32.8	27.3	$20.8x10^{-3}$	33.0	28.6	$23.0x10^{-3}$	34.8	29.8	$24.7x10^{-3}$
160	15	1	32.1	26.9	$20.1x10^{-3}$	31.3	28.8	$24.3x10^{-3}$	34.6	29.5	$24.5x10^{-3}$
		2	33.5	27.6	$21.2x10^{-3}$	32.1	29.4	$25.0x10^{-3}$	35.7	30.3	$25.2x10^{-3}$
	45	1	31.3	26.5	$19.8x10^{-3}$	31.0	28.4	$23.0x10^{-3}$	34.7	29.2	$23.4x10^{-3}$
		2	33.5	27.7	$21.0x10^{-3}$	32.2	29.6	$25.0x10^{-3}$	35.8	30.4	$25.5x10^{-3}$
	60	1	31.4	26.7	$20.3x10^{-3}$	30.6	28.5	$23.9x10^{-3}$	34.6	29.2	$24.4x10^{-3}$
		2	33.4	28.2	$22.0x10^{-3}$	32.3	29.5	$24.0x10^{-3}$	35.6	29.8	$24.7x10^{-3}$
	90	1	30.5	26.5	$20.4x10^{-3}$	30.0	28.4	$23.8x10^{-3}$	32.2	29.7	$24.8x10^{-3}$
		2	32.8	27.6	$214x10^{-3}$	32.0	29.6	$24.6x10^{-3}$	34.8	29.6	$24.6x10^{-3}$

Table 3.2

Psychrometric Data of Moist Air Entering and Leaving the Evaporative Condenser For FD fan with CDE

Supply Voltage	Inclination Angle	No.of stages	Entering Air			Discharge Air Without Heater			Discharge Air with Heater on		
			DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio
V	θ	n	t_1	t_1^*	W_1	t_2	t_2^*	W_2	t_3	t_3^*	W_3
(Volts)	(degree)		(^oC)	(^oC)	(kgw/kgda)	(^oC)	(^oC)	(kgw/kgda)	(^oC)	(^oC)	(kgw/kgda)
230	15	1	29.8	26.9	$21.2x10^{-3}$	30.6	28.7	$24.4x10^{-3}$	34.1	29.7	$24.6x10^{-3}$
		2	29.8	26.9	$21.2x10^{-3}$	32.5	29.0	$25.0x10^{-3}$	35.9	30.0	$25.4x10^{-3}$
	45	1	30.0	26.9	$21.0x10^{-3}$	30.0	28.5	$24.0x10^{-3}$	34.2	20.4	$25.0x10^{-3}$
		2	30.0	26.9	$21.0x10^{-3}$	30.8	28.3	$23.4x10^{-3}$	34.4	29.8	$24.0x10^{-3}$
	60	1	28.8	26.9	$21.6x10^{-3}$	30.1	28.5	$24.0x10^{-3}$	33.7	30.0	$25.6x10^{-3}$
		2	28.8	26.9	$21.6x10^{-3}$	30.6	28.4	$24.6x10^{-3}$	33.5	30.5	$25.0x10^{-3}$
	90	1	29.8	26.8	$21.0x10^{-3}$	30.6	28.8	$24.8x10^{-3}$	32.4	21.0	$26.0x10^{-3}$
		2	29.8	26.8	$21.0x10^{-3}$	31.0	28.8	$24.1x10^{-3}$	32.8	21.0	$26.0x10^{-3}$
	15	1	29.8	26.9	$21.2x10^{-3}$	30.6	28.7	$24.6x10^{-3}$	34.1	29.7	$25.1x10^{-3}$
		2	33.0	27.6	$21.2x10^{-3}$	32.5	29.5	$24.0x10^{-3}$	35.8	30.0	$24.9x10^{-3}$

	45	1	30.0	26.9	21.0×10^{-3}	30.7	28.9	23.4×10^{-3}	34.2	29.4	24.2×10^{-3}
		2	30.4	26.3	20.0×10^{-3}	30.6	28.3	23.4×10^{-3}	34.4	29.1	24.9×10^{-3}
200	60	1	28.8	26.9	21.6×10^{-3}	29.9	28.0	23.7×10^{-3}	34.6	29.9	25.2×10^{-3}
		2	30.7	26.6	20.4×10^{-3}	30.8	28.5	23.8×10^{-3}	34.6	29.7	24.9×10^{-3}
	90	1	29.8	26.8	21.0×10^{-3}	30.5	28.6	24.2×10^{-3}	34.3	30.7	26.0×10^{-3}
		2	31.0	26.9	20.6×10^{-3}	31.2	28.8	24.2×10^{-3}	34.8	30.8	25.8×10^{-3}
	15	1	32.1	26.9	20.2×10^{-3}	31.8	29.0	24.4×10^{-3}	35.0	29.8	24.8×10^{-3}
		2	33.3	27.6	21.2×10^{-3}	32.3	29.0	24.7×10^{-3}	35.9	30.4	25.0×10^{-3}
160	45	1	31.8	26.5	19.6×10^{-3}	31.8	28.5	23.3×10^{-3}	35.0	29.5	24.0×10^{-3}
		2	33.5	27.7	21.0×10^{-3}	32.2	29.4	25.1×10^{-3}	35.9	30.3	25.5×10^{-3}
	60	1	31.4	26.7	20.3×10^{-3}	31.0	28.7	24.2×10^{-3}	34.5	30.5	25.2×10^{-3}
		2	33.4	27.8	21.4×10^{-3}	32.6	29.7	25.0×10^{-3}	26.0	30.5	25.8×10^{-3}
	90	1	30.6	26.6	20.4×10^{-3}	30.4	28.6	24.3×10^{-3}	34.9	29.5	26.0×10^{-3}
		2	32.8	27.3	20.5×10^{-3}	32.6	28.5	25.6×10^{-3}	34.8	29.7	24.8×10^{-3}

Table 3.3

Psychrometric Data of the Moist Air Entering and Leaving the Evaporative Condenser for FD fan with WDE

upply oltage	Inclination Angle	No.of stages	Entering Air			Discharge Air Without Heater			Discharge Air with Heater on		
			DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio
V	θ	n	t_1	t_1^*	W_1	t_2	t_2^*	W_2	t_3	t_3^*	W_3
(Volts)	(degree)		(°C)	(°C)	kgw/kgda)	(°C)	(°C)	kgw/kgda)	(°C)	(°C)	(kgw/kgda)
	15	1	30.0	27.0	20.8×10^{-3}	30.5	28.5	24.0×10^{-3}	32.0	29.0	25.0×10^{-3}
		2	30.0	27.0	20.8×10^{-3}	30.5	28.0	24.0×10^{-3}	31.0	28.5	24.4×10^{-3}
	30	1	29.5	27.0	21.4×10^{-3}	29.6	28.3	23.8×10^{-3}	31.0	29.5	25.1×10^{-3}
		2	29.5	27.2	21.6×10^{-3}	29.8	28.3	23.4×10^{-3}	32.0	29.8	26.0×10^{-3}
	45	1	30.5	27.2	21.4×10^{-3}	30.0	28.5	24.6×10^{-3}	31.0	30.0	26.6×10^{-3}
		2	29.8	27.5	22.2×10^{-3}	29.6	28.6	24.6×10^{-3}	31.5	30.0	26.3×10^{-3}
	60	1	30.3	27.5	22.0×10^{-3}	30.5	29.0	24.8×10^{-3}	31.5	30.5	27.0×10^{-3}
		2	30.0	27.2	21.0×10^{-3}	30.5	29.0	24.8×10^{-3}	31.5	30.3	26.8×10^{-3}
	90	1	30.5	27.0	21.6×10^{-3}	30.8	29.0	24.8×10^{-3}	31.8	30.6	27.4×10^{-3}
		2	30.6	27.3	22.2×10^{-3}	30.5	29.0	24.7×10^{-3}	31.8	30.5	27.3×10^{-3}

Table 3.4

Psychrometric Data of Moist Air Entering and Leaving the Evaporative Condenser for ID fan with WDE

Supply Voltage	Inclination Angle	No.of stages	Entering Air			Discharge Air Without Heater			Discharge Air with Heater on		
			DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio	DBT	WBT	Humidity Ratio
V	θ	n	t_1	t_1^*	W_1	t_2	t_2^*	W_2	t_3	t_3^*	W_3
(volts)	(degree)		(^oC)	(^oC)	(kgw/kgda)	(^oC)	(^oC)	(kgw/kgda)	(^oC)	(^oC)	(kgw/kgda)
230	15	1	32.5	26.9	$20.0x10^{-3}$	32.3	29.0	$24.2x10^{-3}$	35.3	30.6	$24.9x10^{-3}$
		2	32.2	26.6	$19.7x10^{-3}$	32.2	28.9	$23.8x10^{-3}$	35.4	30.1	$25.0x10^{-3}$
	30	1	30.7	27.3	$21.3x10^{-3}$	31.1	29.1	$24.7x10^{-3}$	34.7	30.0	$25.9x10^{-3}$
		2	32.5	27.2	$20.6x10^{-3}$	31.0	28.8	$23.6x10^{-3}$	35.2	30.0	$25.0x10^{-3}$
	45	1	30.5	27.0	$21.0x10^{-3}$	31.0	28.9	$24.4x10^{-3}$	34.5	30.0	$26.4x10^{-3}$
		2	32.0	26.7	$20.0x10^{-3}$	31.9	28.8	$24.0x10^{-3}$	35.0	29.8	$25.9x10^{-3}$
	60	1	29.5	27.3	$22.0x10^{-3}$	30.7	29.1	$24.8x10^{-3}$	34.2	30.0	$26.8x10^{-3}$
		2	31.1	26.8	$20.4x10^{-3}$	31.3	28.7	$24.0x10^{-3}$	35.2	30.2	$26.2x10^{-3}$
	90	1	32.8	26.9	$19.8x10^{-3}$	32.4	28.9	$23.8x10^{-3}$	35.2	30.0	$25.4x10^{-3}$
		2	32.8	26.9	$19.8x10^{-3}$	32.2	28.8	$23.6x10^{-3}$	35.0	30.8	$26.0x10^{-3}$

200	15	1	32.5	26.9	20.2×10^{-3}	32.3	29.0	24.0×10^{-3}	36.6	30.2	24.8×10^{-3}
		2	32.2	29.2	23.4×10^{-3}	32.2	29.1	23.9×10^{-3}	36.3	30.2	25.0×10^{-3}
	30	1	30.8	27.3	21.4×10^{-3}	31.0	29.0	24.6×10^{-3}	34.8	30.3	26.2×10^{-3}
		2	32.5	27.2	20.4×10^{-3}	31.8	29.0	24.3×10^{-3}	35.2	30.0	25.2×10^{-3}
	45	1	30.5	27.4	22.0×10^{-3}	31.3	29.0	24.4×10^{-3}	34.8	30.4	25.6×10^{-3}
		2	32.0	26.7	20.2×10^{-3}	31.8	28.9	24.0×10^{-3}	35.8	29.9	25.6×10^{-3}
	60	1	29.5	27.3	22.4×10^{-3}	31.1	29.1	24.8×10^{-3}	34.5	29.9	26.2×10^{-3}
		2	31.0	26.7	20.6×10^{-3}	31.3	28.9	24.3×10^{-3}	35.0	29.8	26.2×10^{-3}
	90	1	32.5	26.9	20.4×10^{-3}	32.8	28.9	23.6×10^{-3}	34.4	29.9	25.6×10^{-3}
		2	32.5	26.8	20.3×10^{-3}	32.9	28.8	23.4×10^{-3}	34.5	29.8	25.4×10^{-3}
160	15	1	32.5	26.9	20.0×10^{-3}	32.3	29.0	24.2×10^{-3}	36.3	30.2	25.0×10^{-3}
		2	32.2	26.6	19.8×10^{-3}	32.1	28.0	23.2×10^{-3}	36.4	30.1	23.8×10^{-3}
	30	1	30.7	27.3	21.5×10^{-3}	31.0	29.1	24.6×10^{-3}	34.7	30.2	25.4×10^{-3}
		2	32.5	27.2	20.6×10^{-3}	31.5	28.8	24.2×10^{-3}	35.2	30.0	25.0×10^{-3}
	45	1	30.5	27.4	21.8×10^{-3}	31.1	28.9	24.4×10^{-3}	34.5	30.1	25.4×10^{-3}
		2	32.0	26.7	19.8×10^{-3}	31.8	28.8	24.0×10^{-3}	35.9	29.8	24.0×10^{-3}
	60	1	29.5	27.3	22.1×10^{-3}	30.8	28.1	23.8×10^{-3}	34.4	29.8	24.9×10^{-3}
		2	31.1	26.8	20.6×10^{-3}	31.2	28.8	24.3×10^{-3}	34.1	29.8	25.7×10^{-3}
	90	1	32.8	26.9	20.0×10^{-3}	32.5	28.9	23.6×10^{-3}	34.2	29.9	25.3×10^{-3}
		2	32.8	26.9	20.0×10^{-3}	32.5	28.8	23.4×10^{-4}	34.3	29.8	25.0×10^{-3}

Table 3.5

Drift Loss Data for FD fan with CDE

Supply Voltage V Volts	Inclination Angle θ (degree)	No.of stages n	Evaporation loss m_e (kgw/kgda)	Drift Loss m_d (kgw/kgda)
230	15	1	3.3×10^{-3}	0.4×10^{-3}
		2	4.3×10^{-3}	0.3×10^{-3}
	45	1	3.8×10^{-3}	1.0×10^{-3}
		2	4.5×10^{-3}	0.7×10^{-3}
	60	1	4.6×10^{-3}	1.1×10^{-3}
		2	4.2×10^{-3}	1.0×10^{-3}
	90	1	3.4×10^{-3}	2.2×10^{-3}
		2	3.5×10^{-3}	1.9×10^{-3}
200	15	1	4.0×10^{-3}	0.3×10^{-3}
		2	3.8×10^{-3}	0.4×10^{-3}
	45	1	4.0×10^{-3}	0.6×10^{-3}
		2	3.1×10^{-3}	0.6×10^{-3}
	60	1	3.6×10^{-3}	1.2×10^{-3}
		2	3.6×10^{-3}	0.8×10^{-3}
	90	1	3.6×10^{-3}	2.0×10^{-3}
		2	3.7×10^{-3}	1.7×10^{-3}

160	15	1	$4.1x10^{-3}$	$0.2x10^{-3}$
		2	$3.8x10^{-3}$	$0.2x10^{-3}$
	45	1	$3.2x10^{-3}$	$0.4x10^{-3}$
		2	$4.4x10^{-3}$	$0.5x10^{-3}$
	60	1	$3.6x10^{-3}$	$0.5x10^{-3}$
		2	$2.2x10^{-3}$	$0.7x10^{-3}$
	90	1	$3.4x10^{-3}$	$1.0x10^{-3}$
		2	$4.2x10^{-3}$	$1.0x10^{-3}$

Table 3.6

Drift Loss Data for ID fan with CDE

Supply Voltage V Volts	Inclination Angle θ (degree)	No. of stages n	Evaporation loss m_e (kgw/kgda)	Drift Loss m_d (kgw/kgda)
230	15	1	3.2×10^{-3}	0.6×10^{-3}
		2	3.8×10^{-3}	0.4×10^{-3}
	45	1	3.0×10^{-3}	1.0×10^{-3}
		2	2.4×10^{-3}	0.6×10^{-3}
	60	1	2.4×10^{-3}	1.6×10^{-3}
		2	2.0×10^{-3}	1.4×10^{-3}
	90	1	2.8×10^{-3}	2.0×10^{-3}
		2	3.6×10^{-3}	1.8×10^{-3}
200	15	1	3.2×10^{-3}	0.5×10^{-3}
		2	2.8×10^{-3}	0.3×10^{-3}
	45	1	2.4×10^{-3}	0.8×10^{-3}
		2	3.4×10^{-3}	0.5×10^{-3}
	60	1	1.6×10^{-3}	1.3×10^{-3}
		2	3.4×10^{-3}	1.0×10^{-3}
	90	1	3.2×10^{-3}	1.8×10^{-3}
		2	3.6×10^{-3}	1.6×10^{-3}

160	15	1	$4.2x10^{-3}$	$0.4x10^{-3}$
		2	$3.0x10^{-3}$	$0.3x10^{-3}$
	45	1	$3.7x10^{-3}$	$0.7x10^{-3}$
		2	$4.0x10^{-3}$	$0.4x10^{-3}$
	60	1	$3.9x10^{-3}$	$0.4x10^{-3}$
		2	$4.0x10^{-3}$	$1.0x10^{-3}$
	90	1	$3.9x10^{-3}$	$1.7x10^{-3}$
		2	$2.8x10^{-3}$	$1.2x10^{-3}$

Table 3.7

Drift Loss Data for ID fan with WDE

Supply Voltage V Volts	Inclination Angle θ (degree)	No. of stages n	Evaporation loss m_e (kgw/kgda)	Drift Loss m_d (kgw/kgda)
230	15	1	4.2×10^{-3}	1.7×10^{-3}
		2	4.1×10^{-3}	1.2×10^{-3}
	30	1	3.4×10^{-3}	1.2×10^{-3}
		2	3.6×10^{-3}	1.4×10^{-3}
	45	1	4.0×10^{-3}	1.9×10^{-3}
		2	3.4×10^{-3}	2.0×10^{-3}
	60	1	2.8×10^{-3}	2.0×10^{-3}
		2	3.6×10^{-3}	2.2×10^{-3}
	90	1	4.0×10^{-3}	2.6×10^{-3}
		2	3.8×10^{-3}	2.4×10^{-3}
	15	1	3.8×10^{-3}	0.8×10^{-3}
		2	2.4×10^{-3}	1.1×10^{-3}
	30	1	3.2×10^{-3}	1.6×10^{-3}
		2	3.9×10^{-3}	0.9×10^{-3}

200	45	1	2.4×10^{-3}	1.8×10^{-3}
		2	3.8×10^{-3}	1.6×10^{-3}
	60	1	2.4×10^{-3}	1.4×10^{-3}
		2	3.6×10^{-3}	1.4×10^{-3}
	90	1	3.2×10^{-3}	2.0×10^{-3}
		2	3.1×10^{-3}	1.8×10^{-3}
160	15	1	4.2×10^{-3}	0.9×10^{-3}
		2	2.4×10^{-3}	0.6×10^{-3}
	30	1	3.1×10^{-3}	0.8×10^{-3}
		2	3.6×10^{-3}	0.8×10^{-3}
	45	1	2.6×10^{-3}	1.0×10^{-3}
		2	4.2×10^{-3}	1.0×10^{-3}
	60	1	2.7×10^{-3}	1.1×10^{-3}
		2	3.7×10^{-3}	1.4×10^{-3}
	90	1	3.6×10^{-3}	1.7×10^{-3}
		2	3.6×10^{-3}	1.6×10^{-3}

Table 3.8

Drift Loss Data for FD fan with WDE

Supply Voltage V Volts	Inclination Angle θ (degree)	No. of stages n	Evaporation loss m_e (kgw/kgda)	Drift loss m_d (kgw/kgda)
230	15	1	3.2×10^{-3}	1.0×10^{-3}
		2	3.2×10^{-3}	0.4×10^{-3}
	30	1	2.4×10^{-3}	1.4×10^{-3}
		2	2.2×10^{-3}	1.2×10^{-3}
	45	1	3.2×10^{-3}	2.0×10^{-3}
		2	2.4×10^{-3}	1.7×10^{-3}
	60	1	3.8×10^{-3}	2.2×10^{-3}
		2	3.2×10^{-3}	1.9×10^{-3}
	90	1	3.2×10^{-3}	2.6×10^{-3}
		2	2.5×10^{-3}	2.4×10^{-3}

Water flow rate = 60.66 kg/min.

Make-up water = 2.14 kg/min.

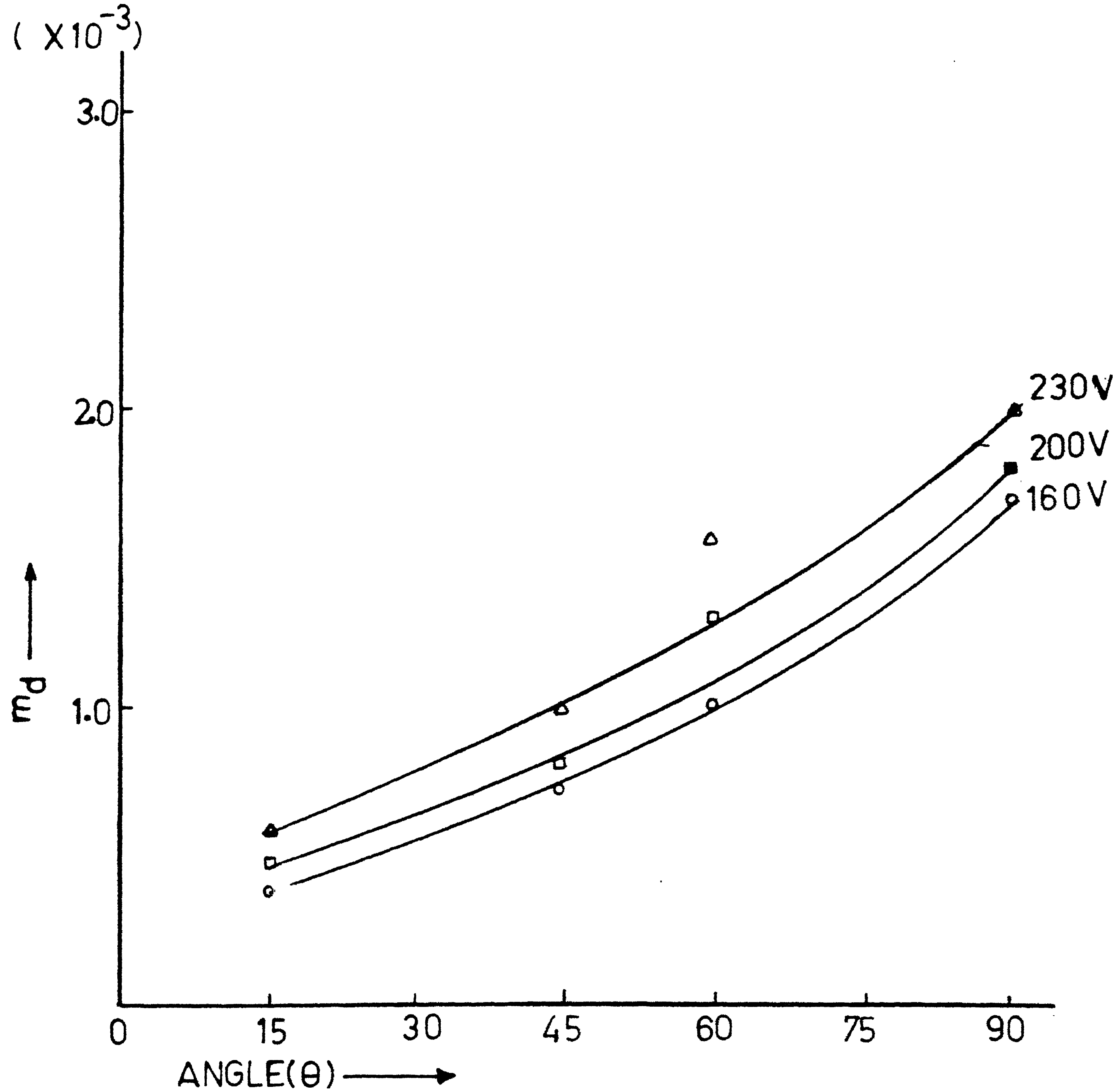

Fig. 3.1

Driftloss vs Inclination angle for ID fan with single stage CDE

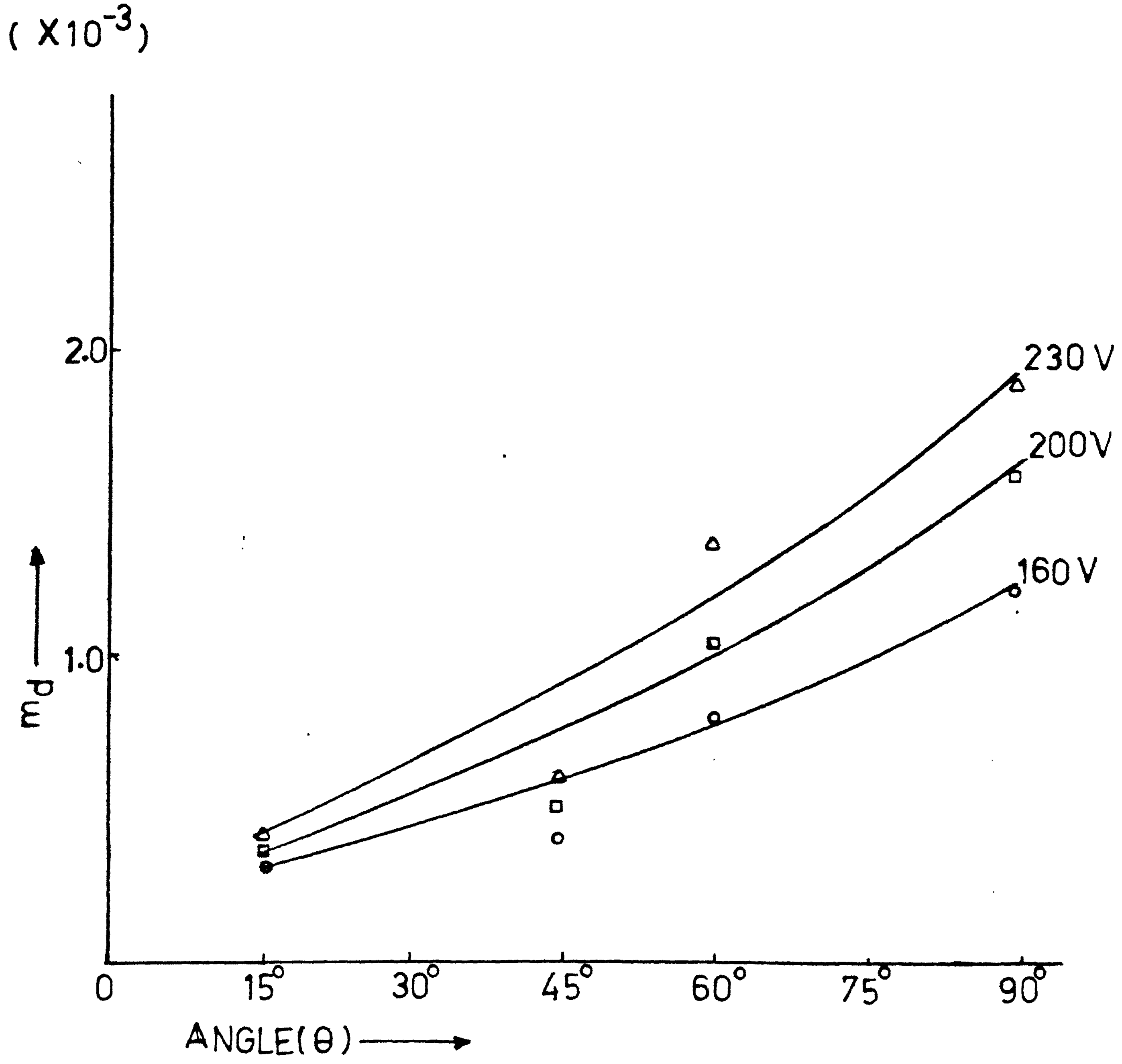

Fig. 3.2

Drift loss vs Inclination angle for ID fan with double stage CDE

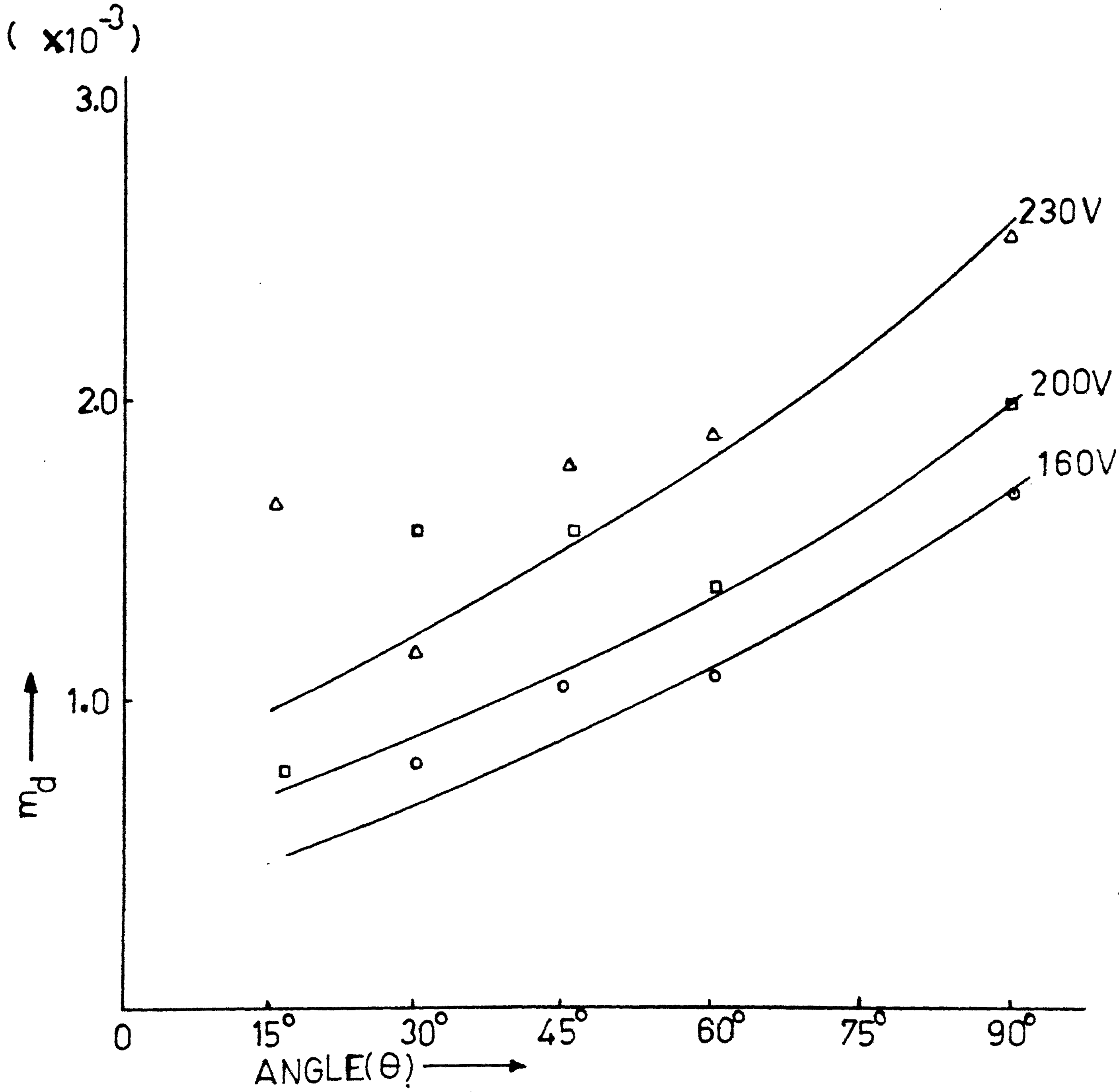

Fig. 3.3

Drift loss vs Inclination angle for ID fan with single stage WDE

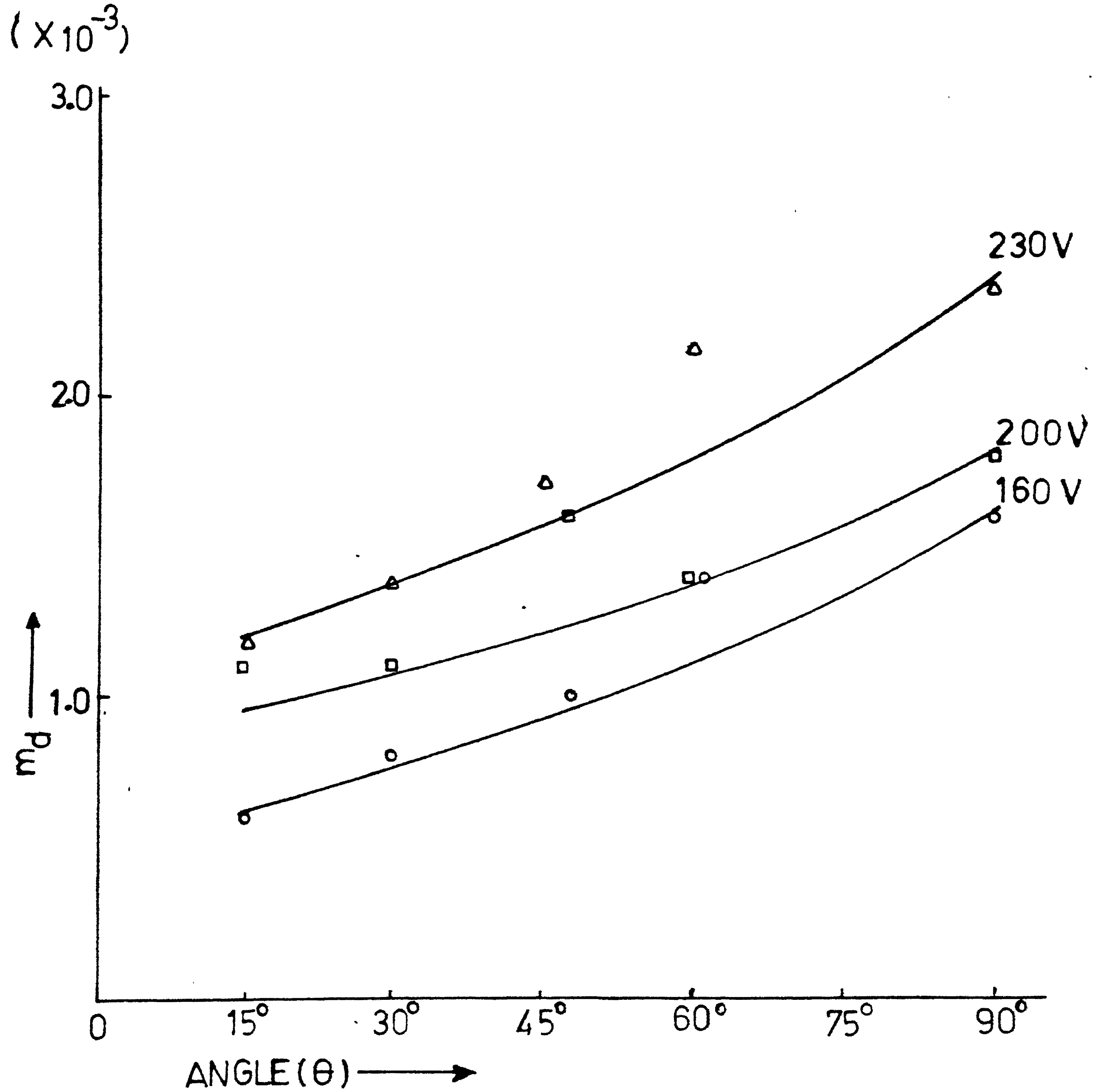

Fig. 3.4

Drift loss vs Inclination angle for ID fan with double stage WDE

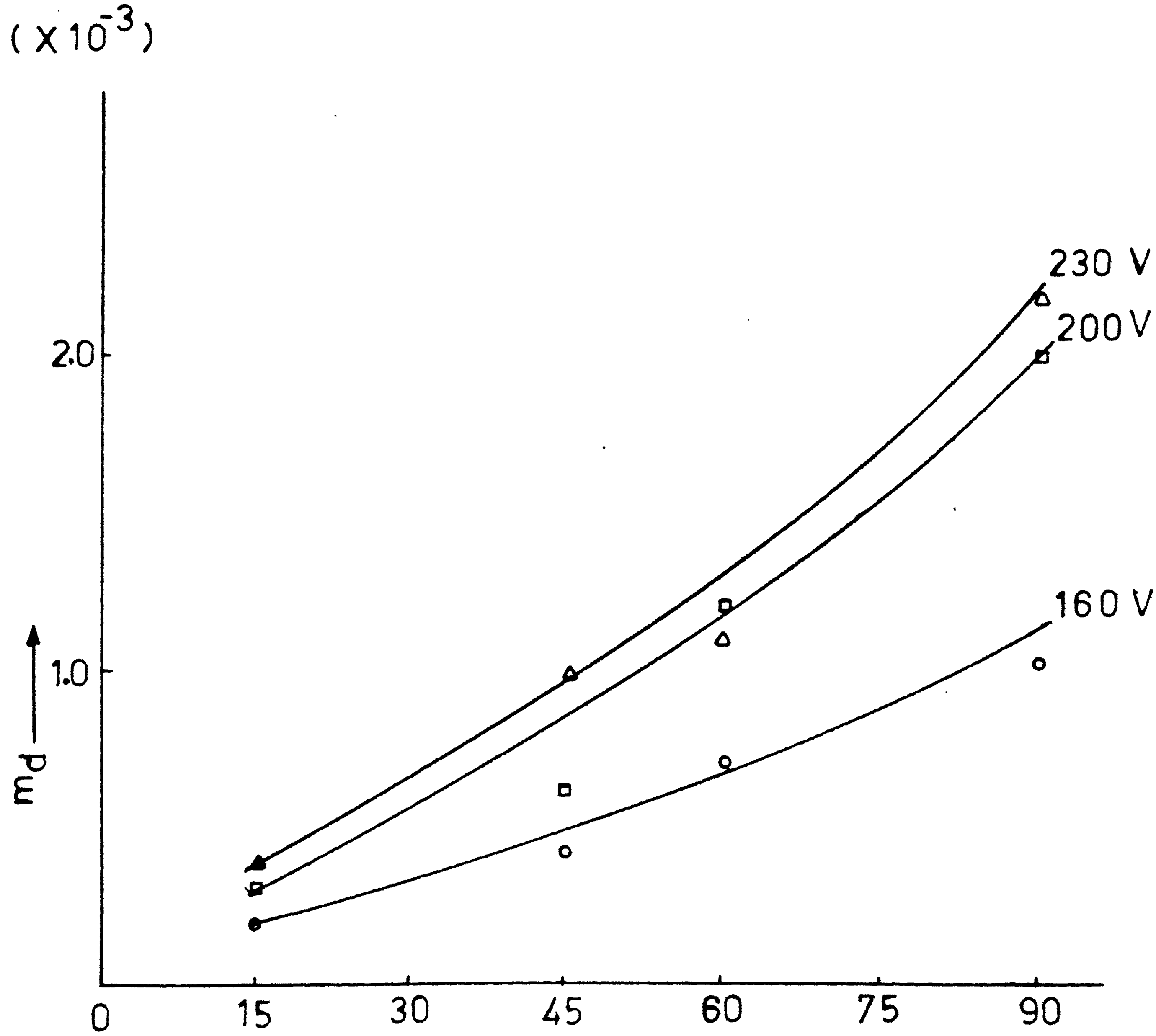

Fig. 3.5

Driftloss vs Inclinatination angle for FDfanwith single stageCDE

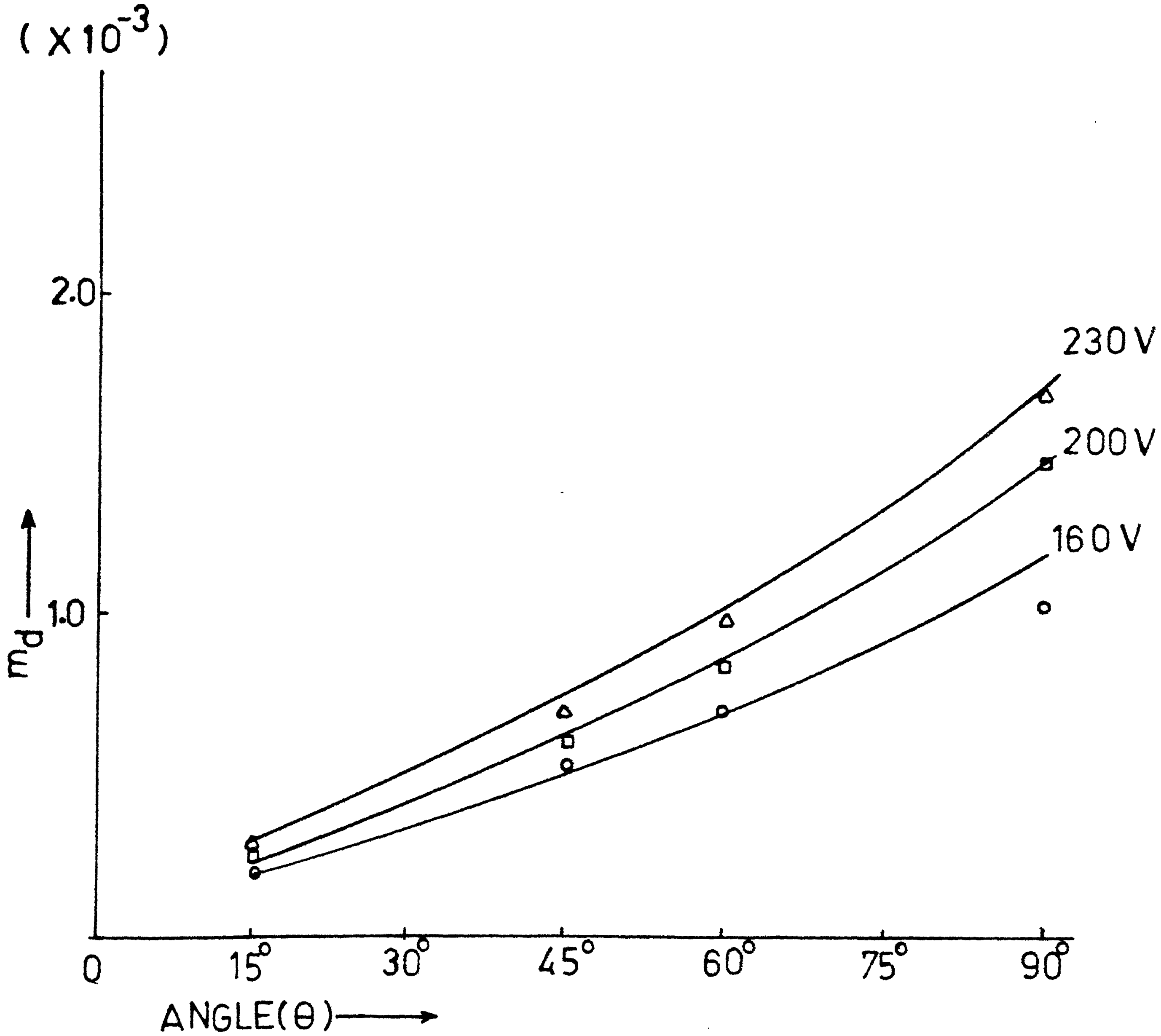

Fig. 3.6

Drift loss vs Inclination angle for FD fan with double stage CDE

eliminators with both FD and ID fans. This difference may be attributed to the larger thickness of the concrete eliminators (25 mm) compared to that of wooden eliminators (13 mm). Air flow rates for various supply voltages are given in appendix.

3.2 PRESSURE DROP:

The pressure drop data were recorded for θ varying between 15^{o} and 90^{o} and for each value of θ, the supply voltage for both ID and FD fan was varied in the range of 160 to 230 Volts A.C. in order to change the speed of the fan and hence the air discharge rate. Single and then double stages for both wooden and concrete drift eliminators were employed first with ID for and then with FD fan. The pressure drops across single stage and double stage were recorded along with the supply voltage and the power drawn by the fan in operation.

The data recorded are shown in Table 3.9 through 3.12. The pressure drop across the various stages of wooden as well as concrete eliminators with ID and FD fan in operation one at a time, are plotted versus θ in Figures 3.7 through 3.14. It can be seen from here that as θ increases, the pressure drop across a particular set of stages decreases with minimum value corresponding to $\theta = 90^{o}$. It can also be seen from Figures 3.8, 3.10, 3.12 and 3.14 that as the number of stages increase, the pressure drop increases which in turn requires a larger amount

Table 3.9

Pressure Drop Data for ID fan with WDE

Inclination Angle θ (degree)	Supply Voltage V (Volts)	No.of stages n	Power (fan) P_{ID} (kW)	Static Pressure P_a (mm of H_2O)	Static Pressure P_b (mm of H_2O)	Static Pressure P_c (mm of H_2O)	Pressure drop ΔP (mm of H_2O)
15	160	1	0.36	8.509	9.652	-	1.143
		2	0.35	7.62	-	9.144	1.524
	180	1	0.37	8.509	9.652	-	1.143
		2	0.37	7.874	-	9.525	1.651
	200	1	0.40	8.382	9.652	-	1.27
		2	0.37	8.128	-	9.652	1.524
	220	1	0.408	8.390	9.652	-	1.262
		2	0.410	8.636	-	9.906	1.27
	230	1	0.46	8.636	10.16	-	1.524
		2	0.425	7.874	-	9.906	2.032

30	160	1	0.33	8.077	8.636	-	0.558
		2	0.325	7.62	-	8.128	0.508
	180	1	0.33	8.00	8.382	-	0.382
		2	0.33	8.128	-	8.636	0.508
	200	1	0.378	8.585	9.144	-	0.558
		2	0.35	7.874	-	8.634	0.762
	220	1	0.38	8.636	9.144	-	0.762
		2	0.375	7.874	-	8.636	0.762
	230	1	0.40	8.636	9.398	-	0.762
		2	0.38	7.874	-	8.89	1.016
45	160	1	0.34	8.128	8.636	-	0.508
		2	0.36	8.128	-	8.89	0.762
	180	1	0.360	8.001	8.509	-	0.508
		2	0.36	7.874	-	8.432	0.558
	200	1	0.365	8.382	8.89	-	0.508
		2	0.40	8.509	-	9.144	0.635

	220	1	0.425	8.128	8.509	-	0.381
		2	0.43	8.509	-	9.144	0.635
	230	1	0.42	8.763	9.144	-	0.381
		2	0.45	8.765	-	9.652	0.889
60	160	1	0.35	8.382	8.636	-	0.254
		2	0.33	8.128	-	8.382	0.254
	180	1	0.345	8.128	8.509	-	0.381
		2	0.35	7.874	-	8.382	0.508
	200	1	0.375	8.636	8.89	-	0.254
		2	0.36	8.128	-	8.636	0.504
	220	1	0.37	8.190	8.509	-	0.319
		2	0.375	8.001	-	8.255	0.254
	230	1	0.425	8.89	9.144	-	0.254
		2	0.40	8.636	-	9.017	0.381

90	160	1	0.36	8.636	8.636	-	0.0
		2	0.35	8.636	-	8.636	0.0
	180	1	0.375	8.636	8.636	-	0.0
		2	0.375	8.636	-	8.636	0.0
	200	1	0.39	8.636	8.636	-	0.0
		2	0.39	8.636	-	8.636	0.0
	220	1	0.42	8.940	8.89	-	0.020
		2	0.42	8.89	-	8.940	0.05
	230	1	0.44	9.144	9.017	-	0.050
		2	0.442	9.144	-	8.89	0.254

Table 3.10

Pressure Drop Data for ID fan with CDE

Inclination Angle	Supply Voltage	No.of stages	Power (fan)	Static Pressure			Pressure drop
θ	V	n	P_{ID}	P_a	P_b	P_c	ΔP
(degree)	(Volts)		(kW)	(mm of H_2O)	(mm of H_2O)	(mm of H_2O)	(mm of H_2O)
15	160	1	0.35	8.255	8.89	-	0.635
		2	0.35	8.128	-	9.144	1.016
	180	1	0.358	8.128	8.703	-	0.635
		2	0.36	7.62	-	9.144	1.524
	200	1	0.375	8.636	9.398	-	0.768
		2	0.375	8.382	-	9.652	1.27
	220	1	0.38	8.128	9.525	-	1.397
		2	0.38	8.001	-	9.652	1.651
	230	1	0.38	8.128	9.271	-	1.143
		2	0.40	8.128	-	9.70	1.574

45	160	1	0.33	8.128	8.636	-	0.508
		2	0.34	7.874	-	8.636	0.762
	180	1	0.35	8.255	8.765	-	0.510
		2	0.35	8.128	-	8.636	0.508
	200	1	0.36	8.255	8.89	-	0.635
		2	0.37	8.128	-	8.89	0.762
	220	1	0.38	8.128	9.017	-	8.889
		2	0.38	8.382	-	9.093	0.711
	230	1	0.40	8.128	9.144	-	0.762
		2	0.405	8.382	-	9.144	1.016
60	160	1	0.35	8.382	8.636	-	0.254
		2	0.34	8.128	-	8.382	0.254
	180	1	0.36	8.128	8.636	-	0.508
		2	0.35	8.128	-	8.585	1.02
	200	1	0.375	8.128	8.636	-	0.508
		2	0.375	8.585	-	8.89	0.3048

	220	1	0.39	8.636	9.144	-	0.508
		2	0.39	8.636	-	9.144	0.685
	230	1	0.41	8.509	8.89	-	0.635
		2	0.42	8.763	-	9.271	0.889
90	160	1	0.37	8.382	8.382	-	0.0
		2	0.34	8.382	-	8.255	0.127
	180	1	0.375	8.636	8.636	-	0.0
		2	0.35	8.128	-	8.509	0.127
	200	1	0.40	8.686	8.89	-	0.127
		2	0.36	8.636	-	8.509	0.127
	220	1	0.41	9.017	9.144	-	0.127
		2	0.375	8.89	-	8.763	0.127
	230	1	0.425	8.763	9.144	-	0.127
		2	0.38	8.89	-	8.763	0.127

Table 3.11

Pressure Drop Data for FD Fan with WDE

Inclination Angle	Supply Voltage	No.of stages	Power (fan)	Static Pressure			Pressure drop
θ	V	n	P_{FD}	P_a	P_b	P_c	ΔP
(degree)	(Volts)		(kW)	(mm of H_2O)	(mm of H_2O)	(mm of H_2O)	(mm of H_2O)
15	160	1	0.44	15.748	13.325	-	2.54
		2	0.44	15.748	-	13.208	2.60
	180	1	0.48	16.256	13.716	-	2.54
		2	0.465	16.256	-	13.716	2.54
	200	1	0.482	16.891	14.097	-	2.794
		2	0.483	16.891	-	13.843	3.048
	220	1	0.475	16.764	13.462	-	3.302
		2	0.476	16.764	-	13.462	3.302
	230	1	0.49	16.256	13.716	-	2.667
		2	0.50	16.51	-	13.462	3.048

30	160	1	0.48	15.494	14.224	-	1.016
		2	0.45	15.748	-	14.732	1.270
	180	1	0.492	16.002	14.732	-	1.27
		2	0.45	15.748	-	14.732	1.016
	200	1	0.495	16.256	15.24	-	1.16
		2	0.45	16.256	-	14.732	1.524
	220	1	0.498	16.256	14.732	-	1.524
		2	0.50	17.018	-	15.748	1.27
	230	1	0.51	16.256	13.716	-	2.54
		2	0.525	17.018	-	14.224	2.794
45	160	1	0.492	15.24	14.732	-	0.508
		2	0.45	15.24	-	14.605	0.635
	180	1	0.490	15.494	14.732	-	0.762
		2	0.46	16.256	-	15.494	0.762
	200	1	0.492	15.24	14.478	-	0.762
		2	0.475	16.51	-	15.494	1.016

	220	1	0.495	15.748	15.24	-	0.508
		2	0.495	15.875	-	15.24	0.635
	230	1	0.495	17.272	16.290	-	1.016
		2	0.52	17.018	-	15.748	1.27
60	160	1	0.485	14.985	14.605	-	0.381
		2	0.492	16.002	-	15.494	0.508
	180	1	0.485	15.494	14.732	-	0.762
		2	0.495	16.256	-	15.748	0.508
	200	1	0.486	14.986	14.859	-	0.127
		2	0.496	16.891	-	16.256	0.635
	220	1	0.492	14.732	14.351	-	0.381
		2	0.52	14.732	-	14.224	0.762
	230	1	0.50	15.24	14.732	-	0.508
		2	0.525	17.272	-	16.51	0.762

90	160	1	0.45	8.636	8.636	-	0.0
		2	0.45	8.636	8.636	-	0.0
	180	1	0.46	8.636	8.636	-	0.0
		2	0.46	8.636	8.636	-	0.0
	200	1	0.47	8.636	8.636	-	0.0
		2	0.47	8.636	8.636	-	0.0
	220	1	0.47	8.636	8.636	-	0.0
		2	0.47	8.639	8.639	-	0.0
	230	1	0.48	9.144	9.017	-	0.127
		2	0.48	9.144	9.017	-	0.127

Table 3.12

Pressure Drop Data for FD fan with CDE

Inclination Angle	Supply Voltage	No.of stages	Power (fan)	Static Pressure			Pressure drop
θ	V	n	P_{FD}	P_a	P_b	P_c	ΔP
(degree)	(Volts)		(kW)	(mm of H_2O)	(mm of H_2O)	(mm of H_2O)	(mm of H_2O)
15	160	1	0.420	14.478	13.716	-	0.889
		2	0.425	14.478	-	12.954	1.524
	180	1	0.425	14.60	13.716	-	0.884
		2	0.43	14.73	-	13.716	1.014
	200	1	0.435	15.24	14.224	-	1.016
		2	0.44	15.24	-	13.20	1.778
	220	1	0.44	15.24	13.208	-	2.032
		2	0.45	15.748	-	13.20	2.54
	230	1	0.470	15.494	14.224	-	1.27
		2	0.475	15.748	-	13.716	2.032

45	160	1	0.425	13.97	13.335	-	0.635
		2	0.425	13.97	-	13.208	0.762
	180	1	0.43	14.224	13.335	-	0.889
		2	0.435	14.732	-	13.97	0.762
	200	1	0.45	15.24	14.35	-	0.889
		2	0.44	15.24	-	14.22	1.016
	220	1	0.45	15.494	14.351	-	1.143
		2	0.46	15.24	-	14.224	1.016
	230	1	0.50	16.00	14.859	-	1.143
		2	0.48	15.24	-	14.224	1.27
60	160	1	0.425	13.87	13.462	-	0.508
		2	0.425	14.22	-	13.589	0.635
	180	1	0.43	14.351	13.58	-	0.762
		2	0.435	15.24	-	14.478	0.762
	200	1	0.46	15.24	14.478	-	0.762
		2	0.44	14.859	-	13.97	0.508

	220	1	0.45	14.732	14.22	-	0.508
		2	0.46	15.24	-	14.478	0.762
	230	1	0.46	14.732	15.29	-	0.508
		2	0.46	14.732	-	13.716	1.016
90	160	1	0.43	15.24	14.986	-	0.254
		2	0.42	13.462	-	13.208	0.254
	180	1	0.42	15.240	14.986	-	0.259
		2	0.425	14.224	-	13.97	0.254
	200	1	0.45	14.986	14.732	-	0.254
		2	0.44	14.478	-	14.224	0.254
	220	1	0.44	14.448	14.224	-	0.254
		2	0.44	14.224	-	14.352	0.127
	230	1	0.475	14.605	14.351	-	0.127
		2	0.46	14.732	-	14.605	0.127

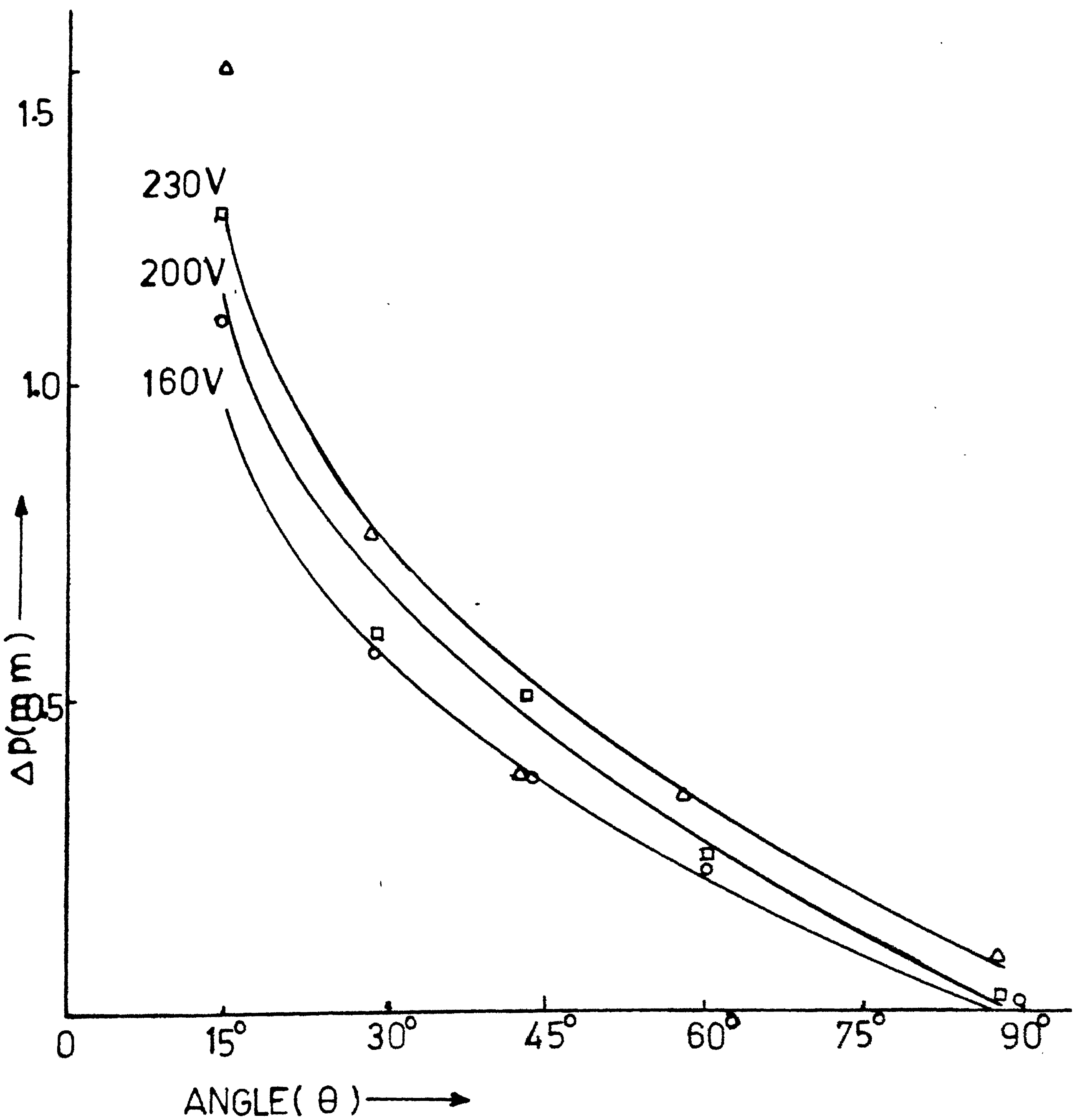

Fig. 3.7

Pressure drop vs inclination angle for ID fan with single stage WDE

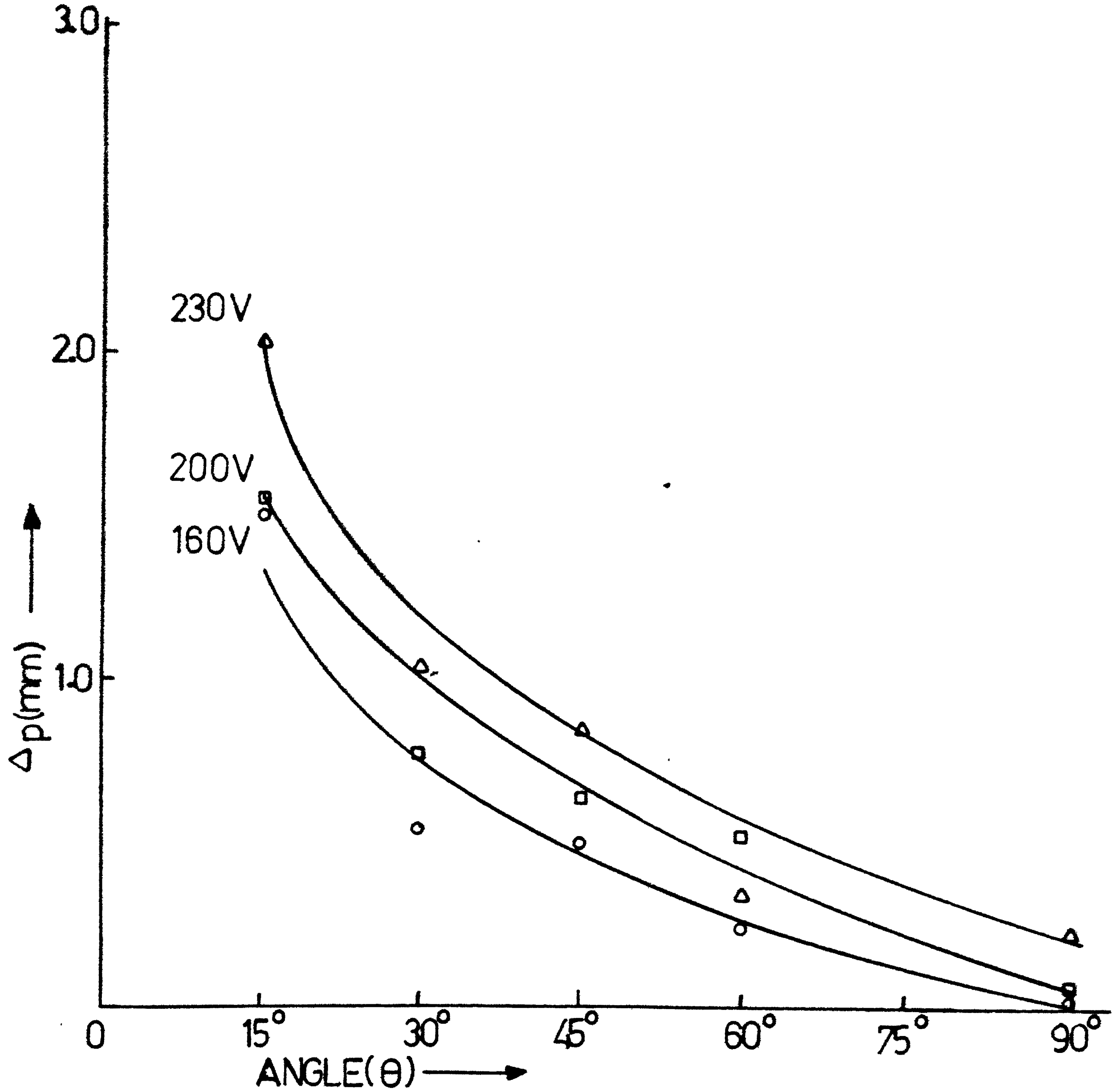

Fig. 3.8

Pressure drop vs Inclination angle for ID fan with double stage WDE

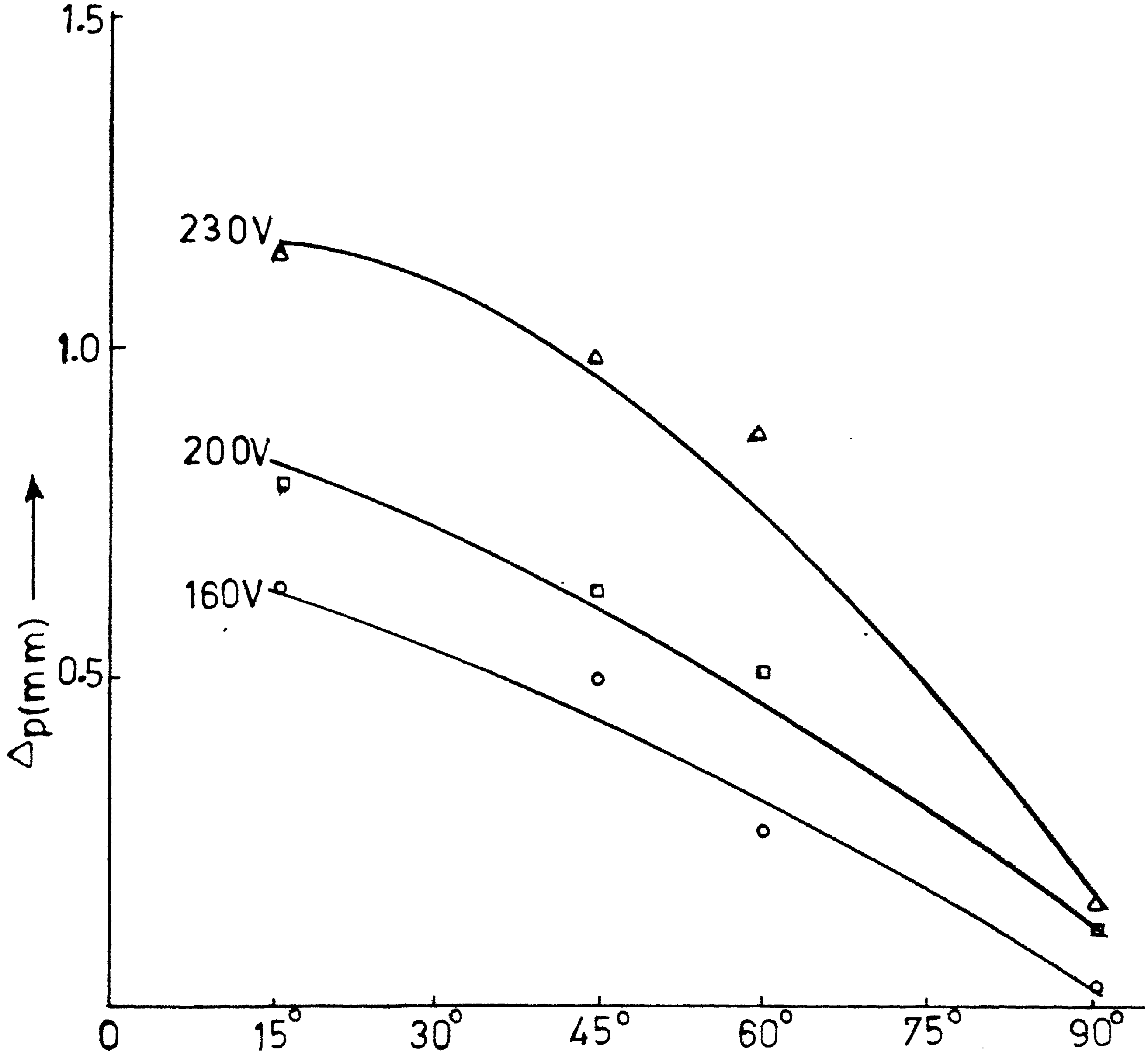

Fig. 3.9

Pressure drop vs Inclination angle for ID fan with single stage CDE

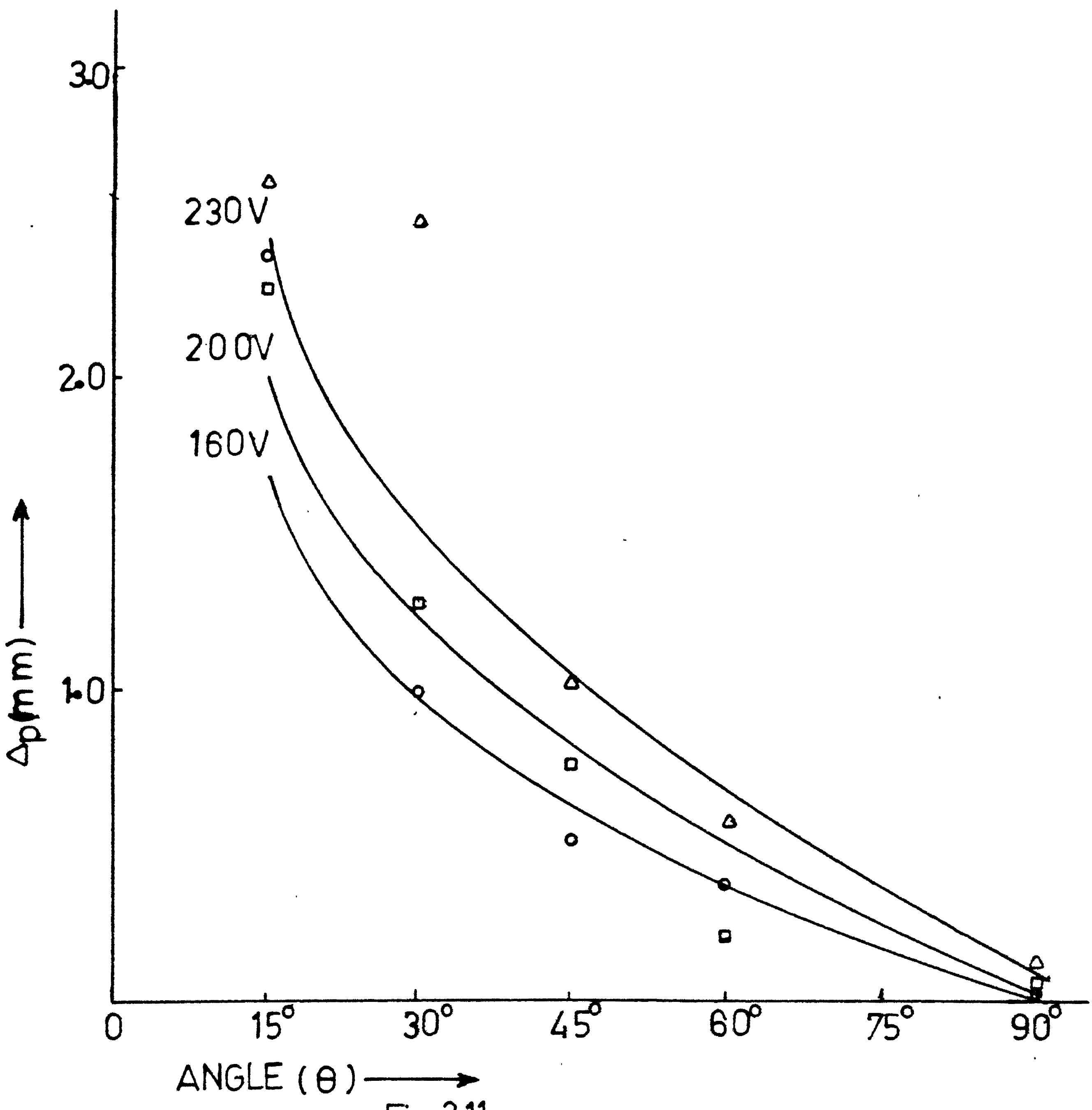

Fig. 3.11

Pressure drop vs Inclination angle for FD fan with single stage WDE

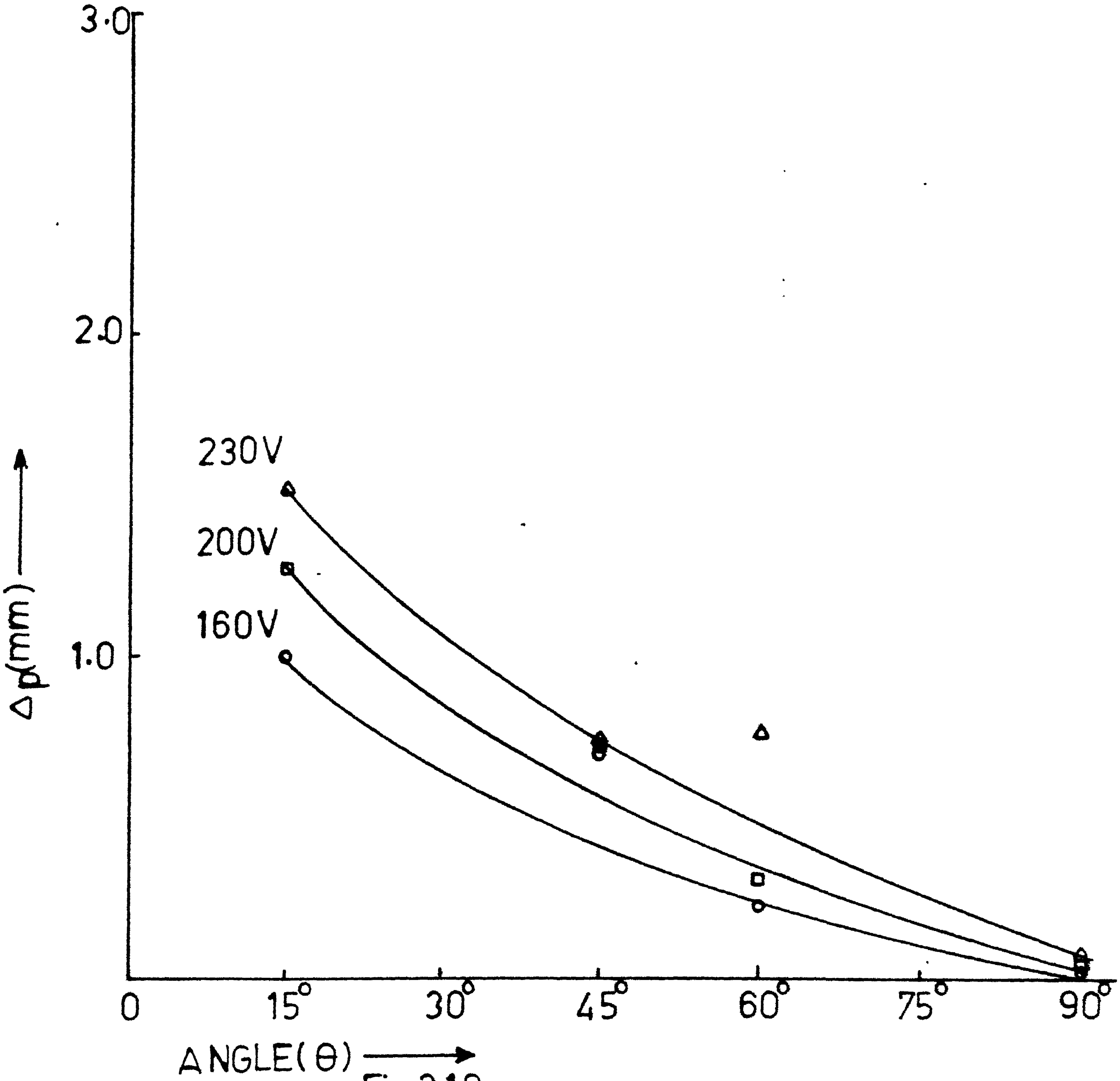

Fig. 3.10

Pressure drop vs Inclination angle for ID fan with double stage CDE

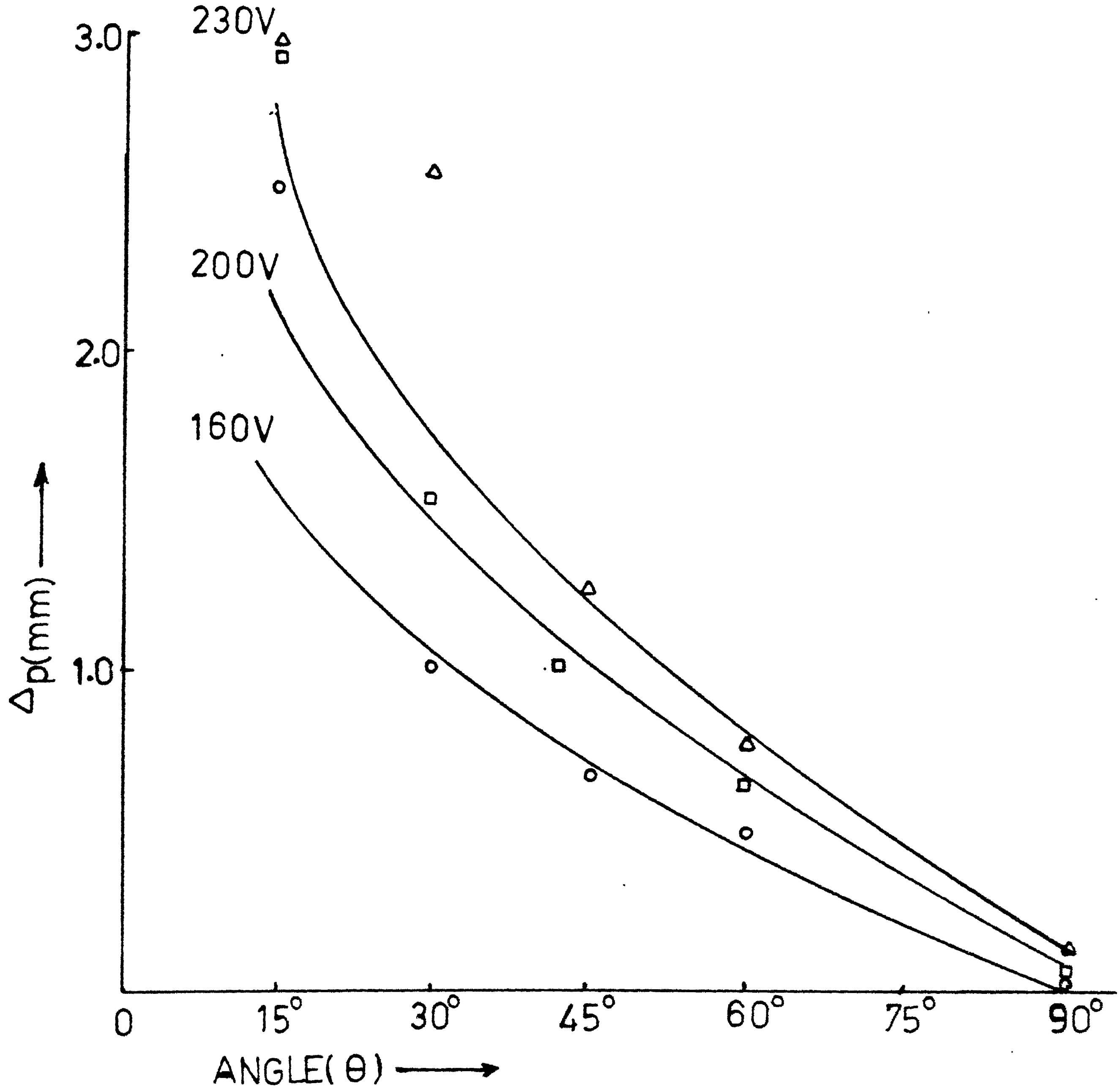

Fig. 3.12

Pressure drop vs Inclination angle for FD fan with double stage WDE

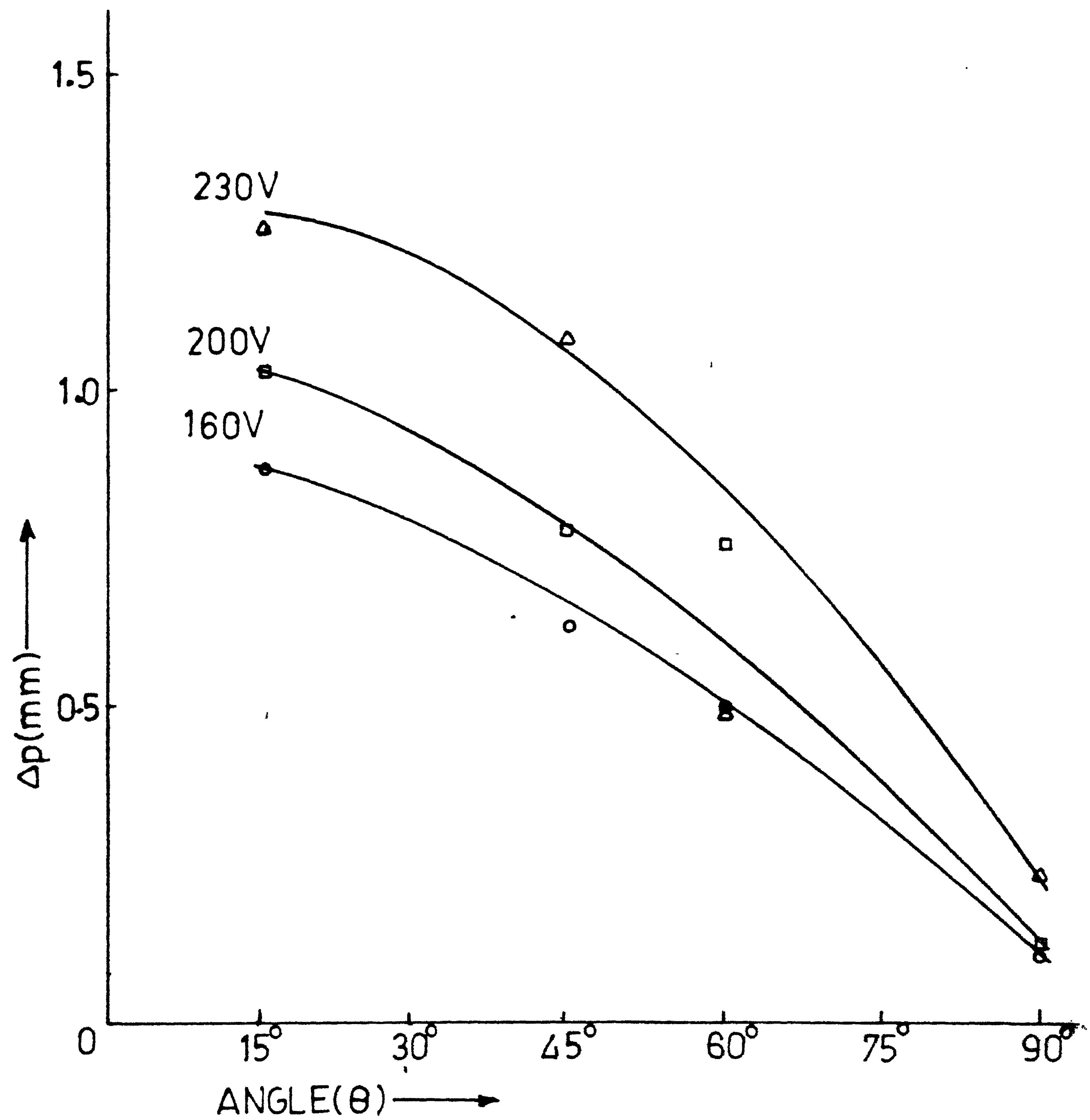

Fig. 3.13

Pressure drop vs Inclination angle for FD fan with singlestage CDE

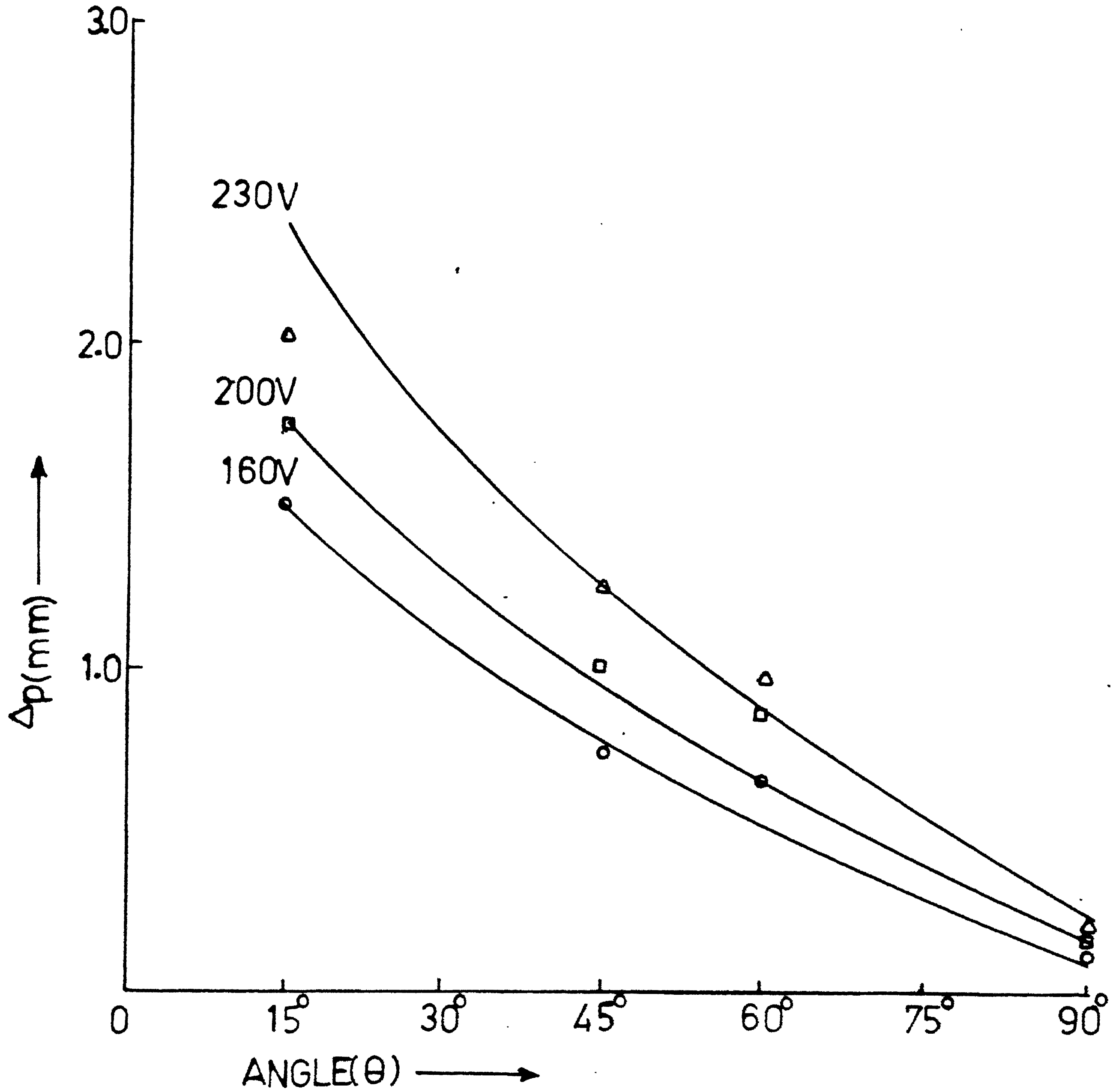

Fig. 3.14

Pressure drop vs Inclination angle for FD fan with double stage CDE

of power input to the fan. It is also noticed that pressure drop is more with wooden eliminators than with concrete drift eliminators. Also the pressure drop across any set of drift eliminators was more with FD fan as compared to ID fan. This may be attributed to significant amount of leakage of air which was inherent in the test rig because of improper sealing of the joints of the ducts resulting in a smaller amount of flow rate through the drift eliminators.

3.3 COP OF THE REFRIGERATION UNIT:

The system performance data are recorded in Tables 3.13 through 3.16 and COP is determined for various cases. The COP is plotted versus θ in Figures 3.15 through 3.22. It can be seen from here that the trend is similar for all the cases and also as the value of θ increases from 15° to 90°, the COP for the system also increases. This is because as the θ increases the volumetric discharge of air through the system increases which results in higher evaporation rate of water thus causing more cooling of water coming in contact with evaporative condenser, thus causing subcooling of the refrigerant and bringing down the condenser pressure. From the figures it is also evident that for lower fan RPM (lower supply voltage) the COP of the system goes down, because of lower air discharge

Table 3.13

Refrigeration System Performance Data for FD fan with WDE

Voltage V (Volts)	Angle of Inclination θ (degree)	No. of stages n	Suction Pressure p_s (kg/cm^2)	Discharge Pressure p_d (kg/cm^2)	Condenser Inlet Temp. T_1 (^oC)	Condenser Outlet Temp. T_2 (^oC)	Power Input Compressor P_{com} (Watts)	COP
230	15	1	3.661	15.845	68.58	32.83	1500	3.876
		2	3.732	15.774	69.0	33.2	1600	3.890
	30	1	3.521	15.492	65.83	29.80	1550	4.059
		2	3.661	15.492	96.08	32.28	1730	2.452
	45	1	3.802	15.915	90.30	33.10	1580	2.673
		2	3.345	15.492	68.0	29.53	1540	3.830
	60	1	3.873	16.126	86.73	33.65	1595	2.866
		2	4.049	15.845	24.43	32.83	1780	2.547
	90	1	3.500	15.674	64.80	29.80	1540	4.115
		2	3.502	15.774	64.90	30.00	1500	4.205

200	15	1	3.732	16.197	80.83	32.83	1590	3.070
		2	3.767	16.338	80.5	33.20	1600	3.160
	30	1	3.697	15.633	78.48	30.0	1575	3.204
		2	4.225	16.408	97.93	30.0	1700	2.601
	45	1	3.943	16.056	92.23	33.65	1580	2.819
		2	3.6619	15.492	84.53	31.73	1640	2.950
	60	1	3.877	15.774	88.93	33.65	1595	2.807
		2	3.943	15.845	91.68	32.83	1690	2.650
	90	1	3.697	15.640	76.48	31.00	1580	3.325
		2	3.710	15.660	78.49	30.39	1670	3.350

160	15	1	4.014	16.197	83.43	32.83	1625	2.949
		2	4.084	16.338	86.20	33.3	1650	2.909
	30	1	3.873	15.845	86.18	29.80	1550	2.867
		2	3.732	16.197	81.23	30.0	1700	3.09
	45	1	3.873	15.845	90.03	33.10	1550	2.688
		2	3.521	15.492	78.48	30.35	1590	3.166
	60	1	3.873	16.056	94.43	33.10	1590	2.506
		2	4.014	15.492	89.48	33.38	1650	2.736
	90	11	3.873	15.492	78.48	30.35	1580	3.202
		2	3.875	15.495	73.60	30.50	1590	3.200

Table 3.14

Refrigeration System Performance Data for FD fan with CDE

Voltage V (Volts)	Angle of Inclination θ (degree)	No. of stages n	Suction Pressure P_s (kg/cm^2)	Discharge Pressure P_d (kg/cm^2)	Condenser Inlet Temp. T_1 (^oC)	Condenser Outlet Temp. T_2 (^oC)	Power Input Compressor P_{com} (Watts)	COP
230	15	1	4.084	16.549	97.18	29.53	1660	2.478
		2	4.30	16.971	99.38	33.38	1610	2.414
	45	1	4.225	16.690	101.58	32.28	1750	2.342
		2	4.295	16.971	97.18	32.83	1640	2.486
	60	1	4.295	16.760	106.53	32.83	1750	2.222
		2	4.366	16.901	92.78	32.28	1610	2.700
	90	1	4.084	16.549	93.88	30.63	1600	2.617
		2	3.767	16.197	82.33	29.80	1550	3.053

200	15	1	4.084	16.760	101.03	29.8	1650	1.962
		2	4.225	16.971	103.7	33.10	1600	1.964
	45	1	4.295	16.830	109.8	31.73	1770	2.122
		2	4.260	17.042	104.88	32.83	1650	1.965
	60	1	4.225	16.830	109.2	32.28	1720	2.083
		2	4.295	17.042	104.33	32.55	1650	1.968
	90	1	4.154	16.760	102.13	30.90	1620	2.328
		2	3.943	16.338	90.58	30.0	1560	2.677
160	15	1	4.225	16.549	105.43	30.63	1700	2.231
		2	4.436	17.042	104.8	33.38	1650	2.088
	45	1	4.225	16.830	108.18	32.28	1750	2.134
		2	4.154	17.042	106.5	32.83	1650	2.194
	60	1	4.366	16.901	109.83	31.73	1710	1.963
		2	4.225	16.971	105.70	32.28	1650	2.230
	90	1	4.084	16.549	105.40	30.63	1600	2.226
		2	4.049	16.619	98.28	30.63	1600	2.416

Table 3.15

Refrigeration System Performance Data for ID fan with WDE

Voltage V (Volts)	Angle of Inclination θ (degree)	No. of stages n	Suction Pressure P_s (kg/cm^2)	Discharge Pressure P_d (kg/cm^2)	Condenser Inlet Temp. T_1 (^oC)	Condenser Outlet Temp. T_2 (^oC)	Power Input Compressor P_{com} (Watts)	COP
230	15	1	4.507	17.464	114.78	32.83	2020	2.010
		2	4.401	17.464	111.48	34.75	1950	2.081
	30	1	4.084	17.253	98.83	28.98	1700	2.416
		2	4.366	17.394	104.88	31.18	1700	2.262
	45	1	4.401	17.429	111.48	30.63	1780	2.076
		2	4.401	17.429	110.93	34.48	1900	2.082
	60	1	4.436	17.429	110.93	31.73	1800	2.286
		2	4.330	17.253	101.03	32.28	1700	2.354
	90	1	4.401	17.253	92.23	33.54	1900	2.701
		2	4.415	17.254	92.3	33.54	1880	2.700

200	15	1	4.330	17.3943	116.98	32.55	1950	1.944
		2	4.225	17.3239	117.53	33.10	1915	1.906
	30	1	4.366	17.394	103.23	29.53	1720	2.279
		2	4.401	17.394	110.93	30.63	1700	2.087
	45	1	4.401	17.394	113.13	30.63	1730	2.026
		2	4.225	17.323	115.8	33.93	1820	1.954
	60	1	4.225	17.323	113.13	31.18	1810	2.010
		2	4.401	17.253	107.08	32.0	1700	2.189
	90	1	4.401	17.253	105.4	30.0	1950	2.356
		2	4.403	17.254	104.5	30.5	1940	2.340

160	15	1	4.330	17.394	117.8	32.0	1985	1.906
		2	4.577	17.535	117.5	34.2	1950	1.931
	30	1	4.401	17.394	107.0	29.25	1720	2.192
		2	4.401	17.394	112.30	30.63	1700	2.057
	45	1	4.429	17.464	113.68	31.45	1725	2.032
		2	4.788	17.429	116.70	35.90	1800	2.000
	60	1	4.401	17.429	113.68	31.18	1810	2.027
		2	4.366	17.394	109.2	31.18	1695	2.119
	90	1	4.577	17.429	115.8	30.08	1990	2.001
		2	4.560	17.5	116.8	30.00	2000	2.040

Table 3.16

Refrigeration System Performance Data for ID fan with CDE

Voltage V (Volts)	Angle of Inclination θ (degree)	No. of stages n	Suction Pressure P_s (kg/cm^2)	Discharge Pressure P_d (kg/cm^2)	Condenser Inlet Temp. T_1 (^oC)	Condenser Outlet Temp. T_2 (^oC)	Power Input Compressor P_{com} (Watts)	COP
230	15	1	4.401	16.830	112.0	30.08	1680	2.471
		2	4.154	16.901	96.63	30.90	1600	2.518
	45	1	4.366	17.042	113.68	33.38	1800	2.052
		2	4.295	16.971	109.83	31.73	1690	2.120
	60	1	4.401	16.901	108.73	31.73	1800	2.151
		2	4.154	16.971	100.5	30.63	1700	2.357
	90	1	4.401	17.147	114.23	33.65	1950	2.037
		2	4.401	17.253	105.43	33.93	1670	2.245

200	15	1	4.401	17.147	114.23	33.65	1950	2.023
		2	4.366	17.323	109.55	33.38	1660	2.008
	45	1	4.507	17.112	114.78	33.38	1790	2.035
		2	4.084	16.901	111.7	31.73	1660	2.331
	60	1	4.401	17.112	112.0	31.45	1810	2.067
		2	4.154	16.901	108.18	30.35	1680	2.132
	90	1	4.084	16.549	99.93	30.08	1700	2.386
		2	4.154	16.901	96.63	30.90	1600	2.510
160	15	1	4.436	17.112	117.53	33.93	1950	1.956
		2	4.401	17.464	112.03	33.38	1710	2.066
	45	1	4.436	17.077	115.33	32.28	1800	2.002
		2	4.436	17.183	113.13	32.0	1700	2.034
	60	1	4.295	17.042	115.88	31.73	1950	1.989
		2	4.436	17.112	111.48	30.35	1700	2.085
	90	1	4.225	16.901	111.48	30.35	1715	2.115
		2	4.40	17.112	109.83	31.18	1675	2.109

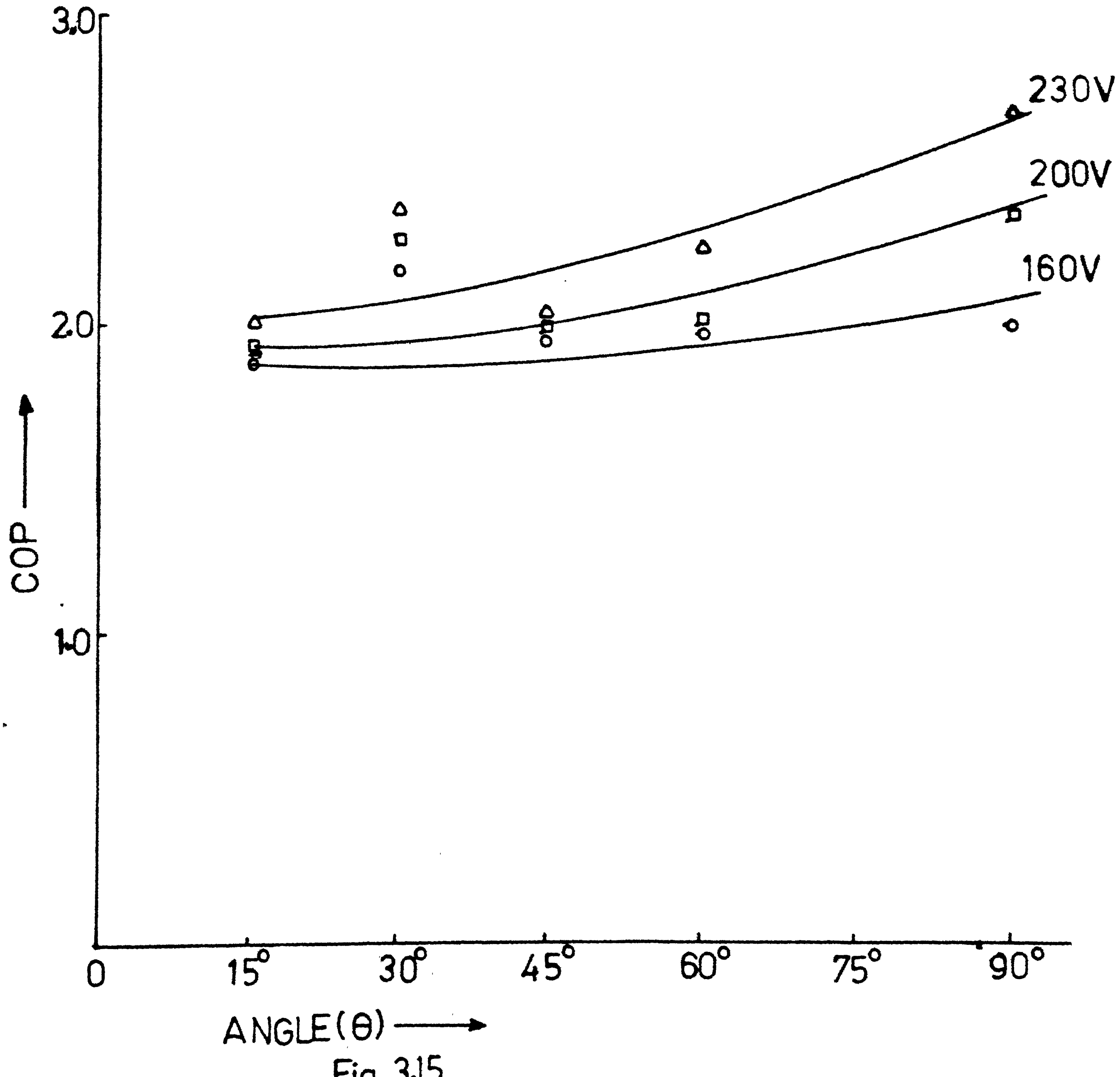

Fig. 3.15

COP vs Inclination angle for ID fan with single stage WDE

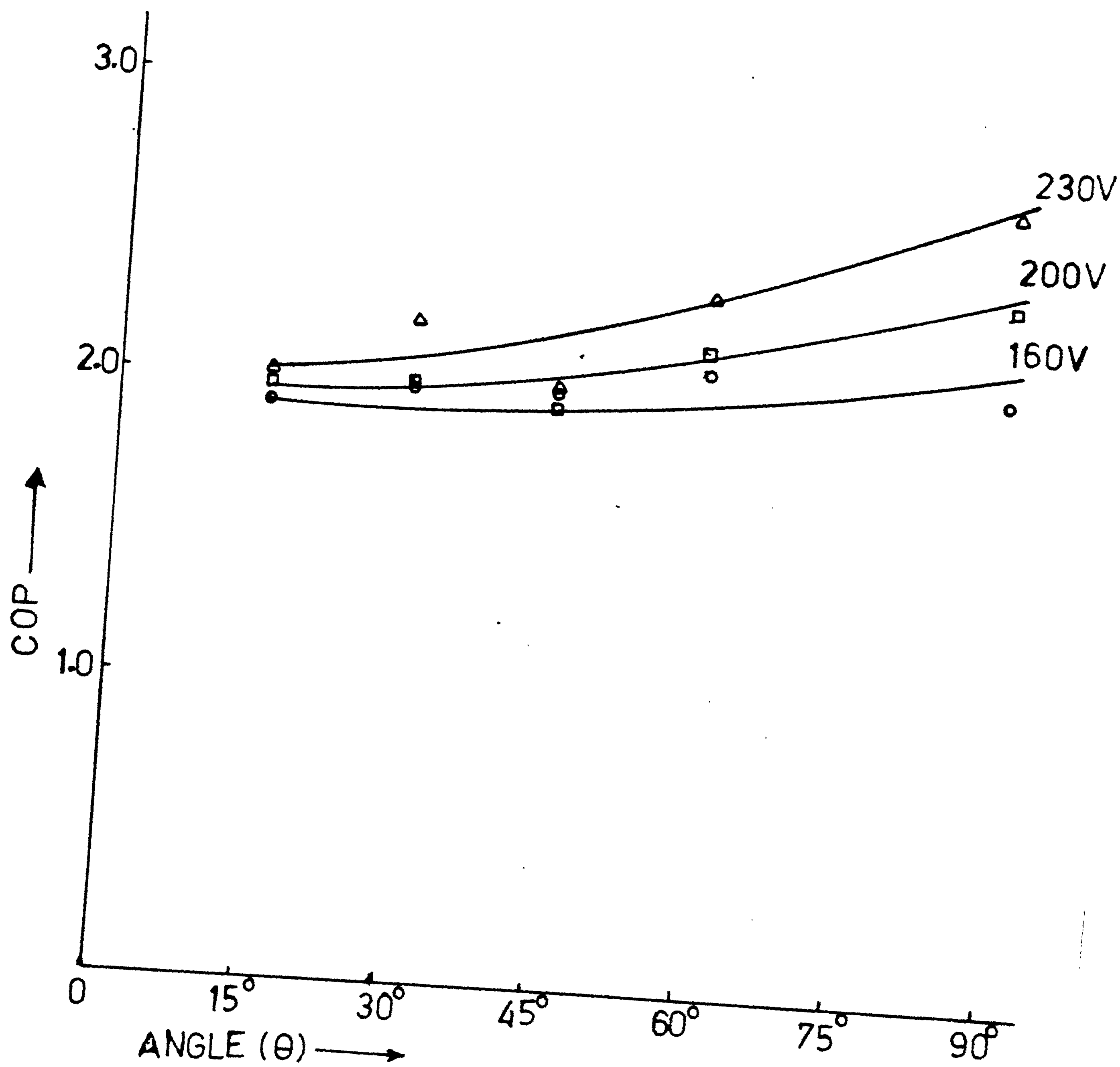

Fig. 3.16

COP vs Inclination angle for ID fan with double stage WDE

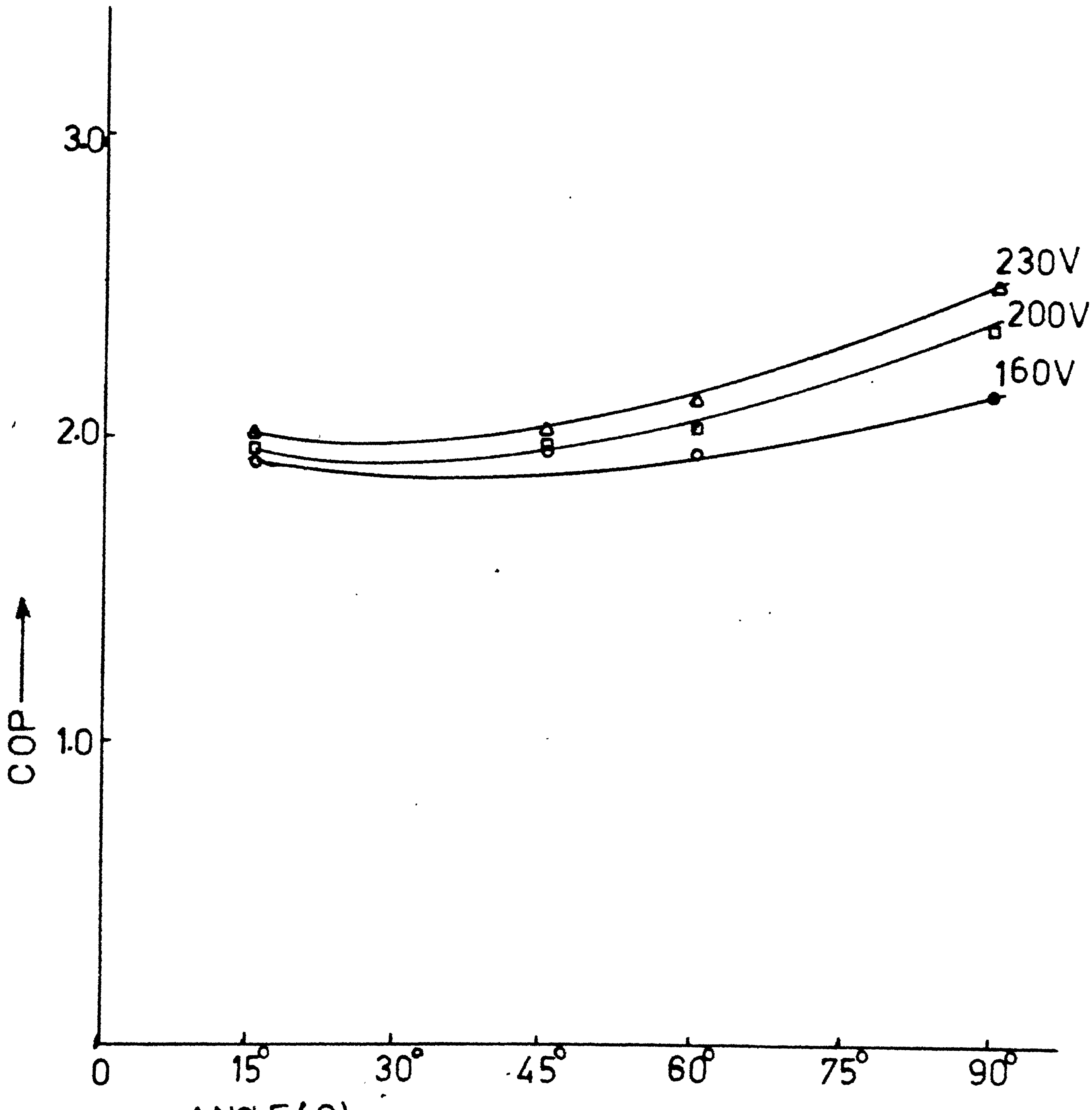

Fig. 3.17

COP vs Inclination angle for ID fan with single stage CDE

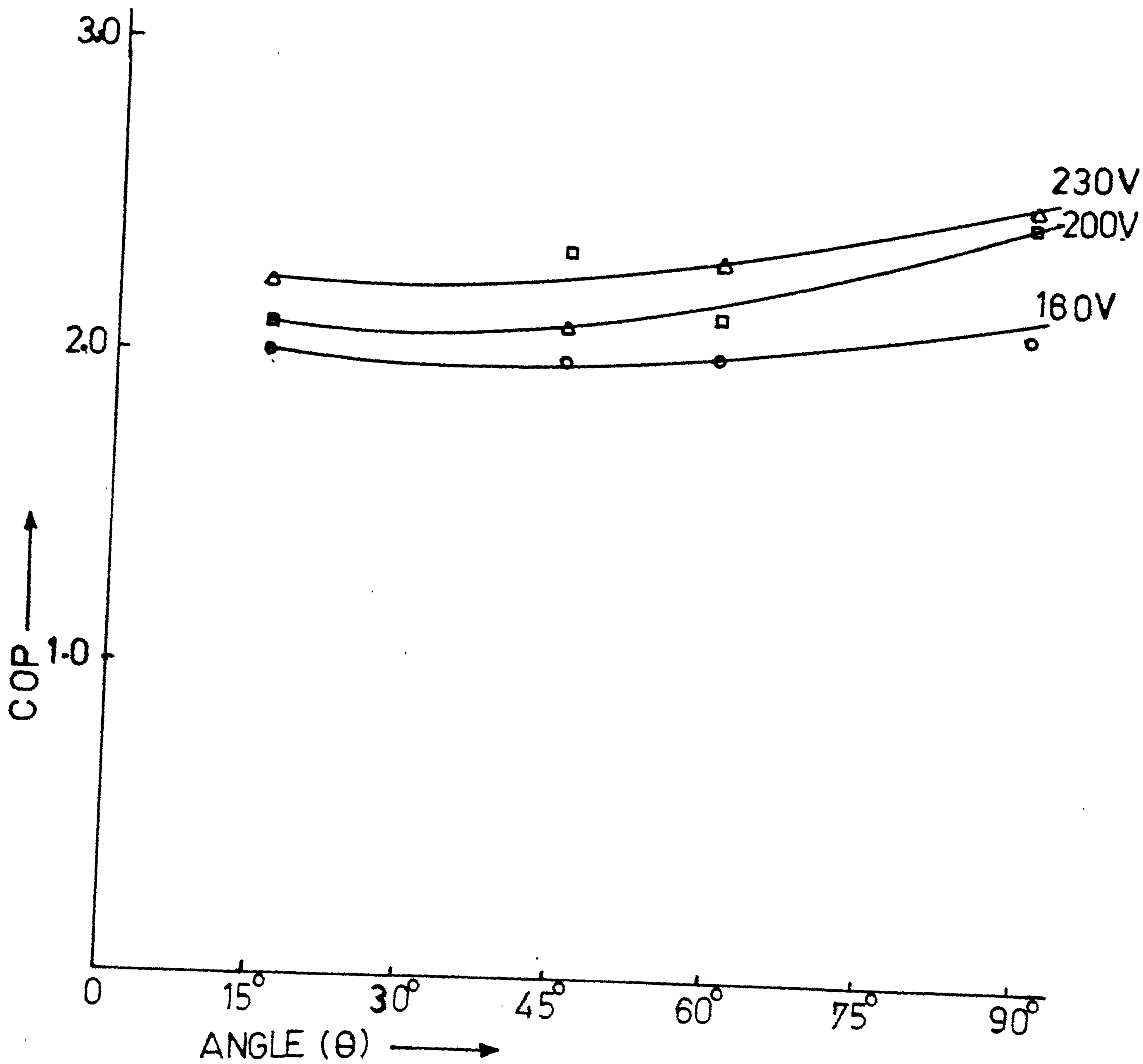

Fig. 3.18

COP vs Inclination angle for ID fan with double stage CDE

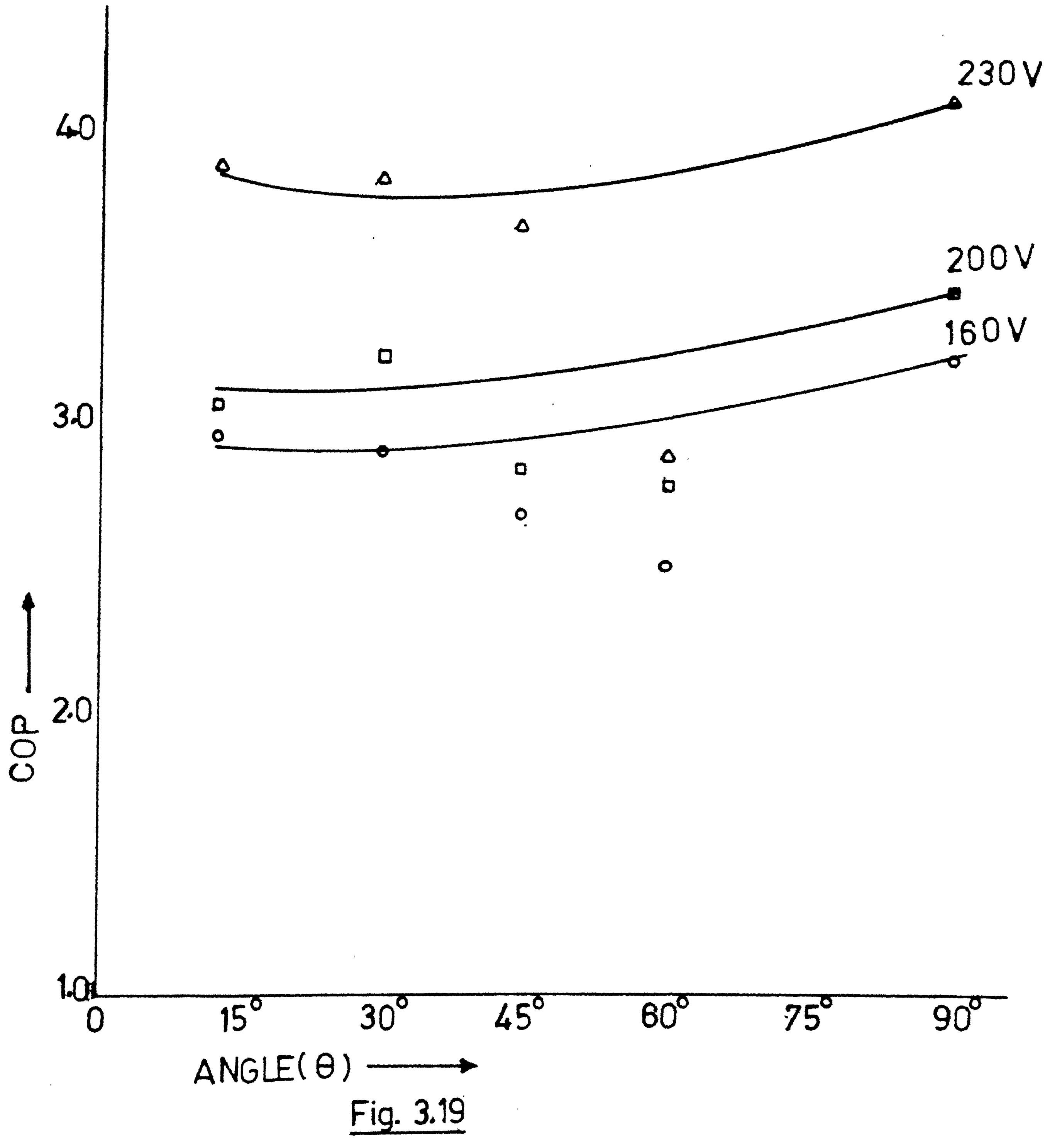

Fig. 3.19

COP vs Inclination angle for FD fan with single stage WDE

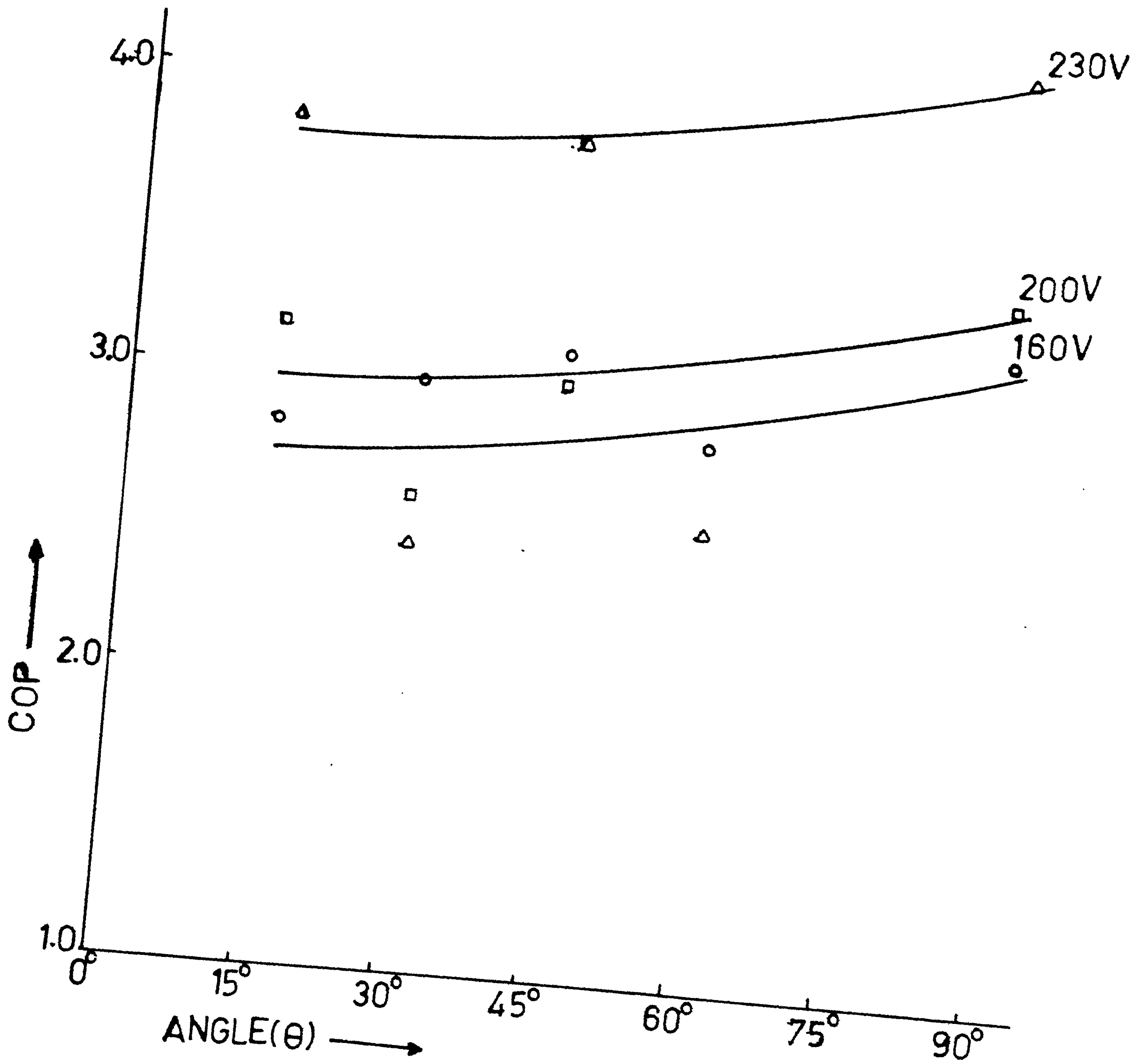

Fig. 3.20

COP vs Inclination angle for FD fan with double stage WDE

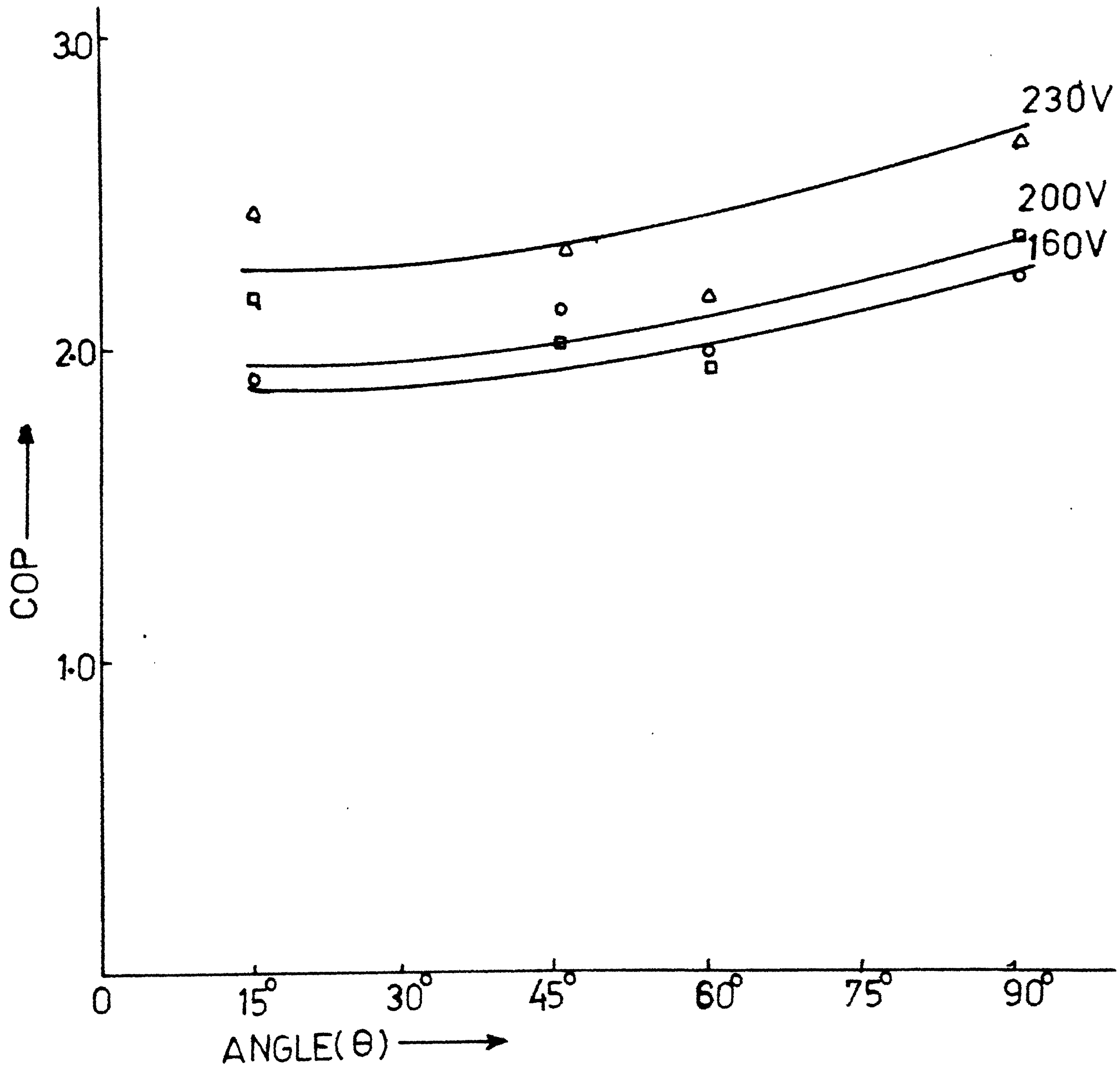

Fig. 3.21

COP vs Inclination angle for FD fan with single stage CDE

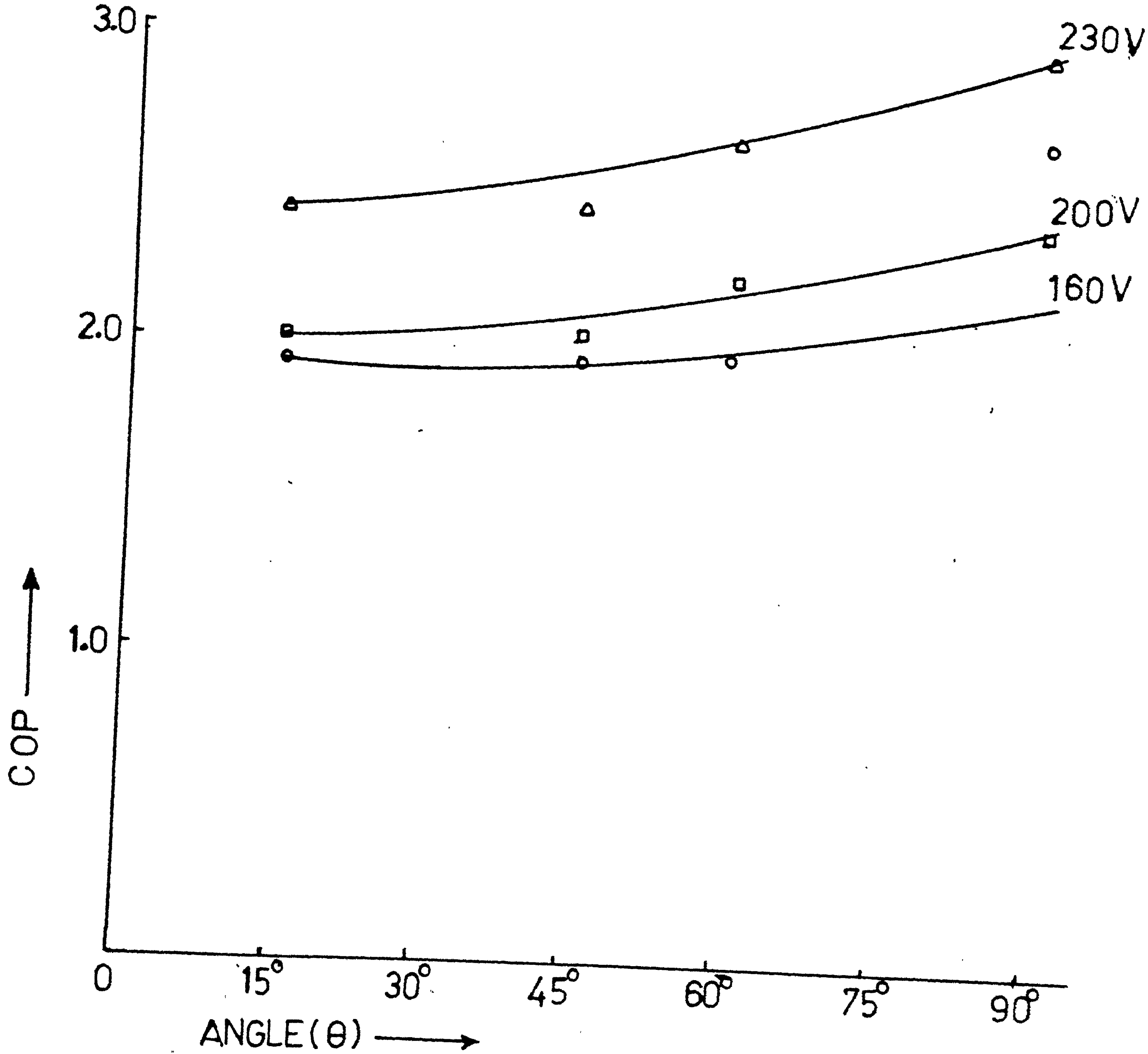

Fig. 3.22

COP vs Inclination angle for FD fan with double stage CDE

rate of the fan. Also when the number of stages increases, the net effect is a drop in COP due to increased resistance to flow and resulting lower air discharge rate. Comparative study of Figures 3.15 through 3.18 shows that COP of the system is more with FD fan than with ID fan for various cases. This can be explained on the basis at greater degree of turbulence is created by FD fan in the evaporative condenser box as compared with ID fan and this leads to higher rate of heat transfer from the condenser coils.

3.4 OPTIMUM ANGLE OF ORIENTATION:

For determination of an optimum angle of orientation for drift eliminators, running cost analysis was carried out. The main factors considered were power loss due to pressure drop across the drift eliminators and the amount of water lost to the atmosphere in the form of drift. The cost of water (for industrial purposes) was taken Rs. 1.25 per 1000 litres and the cost of power per kWhr was taken Rs. 0.70 for the purpose of cost analysis.

Initially, drift loss was calculated in terms of kg per hour and then multiplied by the cost per litre of water. This gave the cost of drift loss (C_1) in terms of 'paise'.

Similarly, the power loss due to pressure drop across the drift eliminators was found out in terms of 'paise' per hour. The method of calculating power loss due to pressure drop is shown below.

Power loss due to pressure drop

$$= \Delta p \times \text{Volumetric discharge per hour } (V_{vol})$$

$$= (\rho_w g h) \times V_{vol} \qquad (3.1)$$

Power lost in terms of cost, C_2 (paise)

$$= \frac{(\rho_w g h \times V_{vol}) \times 70 \times 10^{-3}}{3.6 \times 10^{6}} \qquad (3.2)$$

Where,

ρ_w = density of water, kg/m^3

g = acceleration due to gravity (9.80, m/s^2)

h = pressure drop in mm of water

V_{vol} = Volumetric discharge of air, m^3/h

Equation (3.2) gives the cost of power lost to overcome the pressure drop across the drift eliminators. The values of

Table 3.17

Drift Loss and Pressure Drop Costs for FD fan with single stage CDE

Supply Voltage = 230 V AC
Entering Air velocity = 11.5 m/s

Inclination Angle θ (degree)	Specific Drift Loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (costs of water lost (paise)	C_2 (Costs of pressure drop (paise)
15	0.4×10^{-3}	1.27	1.892	0.2365	1.019
45	1.0×10^{-3}	1.143	4.732	0.5915	0.917
60	1.1×10^{-3}	0.508	5.205	0.650	0.4077
90	2.2×10^{-3}	0.127	10.41	1.30	0.1019

Table 3.18

Drift Loss and Pressure drop Costs for ID fan with single stage CDE

Supply Voltage = 230 V (AC)

Entering air velocity = 7.5 m/s

Inclination Angle θ (degree)	Specific Drift loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (costs of water lost (paise)	C_2 (Costs of Pressure drop (paise)
15	0.4×10^{-3}	1.143	1.236	0.1545	0.5978
45	1.0×10^{-3}	1.016	3.091	0.3863	0.5314
60	1.6×10^{-3}	0.889	4.945	0.6181	0.4650
90	2.0×10^{-3}	0.127	6.182	0.7727	0.0664

Table 3.19

Drift Loss and Pressure Drop Costs for ID fan with single stage WDE

Supply Voltage = 230 V AC
Entering Air Velocity = 7.66 m/s

ion Angle θ (degree)	Specific Drift Loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (Costs of water lost (paise)	C_2 (Costs of pressure drop (paise)
15	1.0×10^{-3}	1.524	3.145	0.393	0.8147
30	1.2×10^{-3}	0.762	3.774	0.471	0.4073
45	1.9×10^{-3}	0.381	5.975	0.746	0.2036
60	2.0×10^{-3}	0.254	6.290	0.786	0.1357
90	2.6×10^{-3}	0.050	8.177	1.022	0.026

Table 3.20

Drift Loss and Pressure Drop costs for FD fan with double stage CDE

Supply Voltage = 230 V AC

Entering Air Velocity = 113.0 m/s

Inclination Angle θ (degree)	Specific Drift loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (Costs of water lost (paise)	C_2 (Costs of pressure drop (paise)
15	0.3×10^{-3}	2.032	1.395	0.1743	1.6025
45	0.7×10^{-3}	1.270	3.255	0.4068	1.0015
60	1.0×10^{-3}	1.016	4.650	0.58125	0.7591
90	1.9×10^{-3}	0.127	8.835	1.1043	0.1001

Table 3.21

Drift Loss and Pressure Drop Costs for ID fan with double stages CDE

Supply Voltage = 230 V AC

Entering Air Velocity = 7.46 m/s

Inclination Angle θ (degree)	Specific Drift loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (Costs of water lost (paise)	C_2 (Costs of pressure drop (paise)
15	0.4×10^{-3}	1.524	1.230	0.1537	0.893
45	0.6×10^{-3}	1.016	1.846	0.2307	0.5289
60	1.4×10^{-3}	0.889	4.307	0.5383	0.3628
90	1.8×10^{-3}	0.127	5.538	0.692	0.0661

Table 3.22

Drift Loss and Pressure Drop Costs for ID fan with double stages WDE

Supply Voltage = 230 V AC

Entering Air Velocity = 7.45 m/s

Inclination Angle θ (degree)	Specific Drift loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (Costs of water lost (Paise)	C_2 (Costs of pressure drop (paise)
15	1.2×10^{-3}	2.032	3.670	0.458	1.056
30	1.4×10^{-3}	1.016	4.282	0.535	0.528
45	1.9×10^{-3}	0.888	5.811	0.7263	0.4622
60	2.2×10^{-3}	0.381	6.729	0.8411	0.1980
90	2.4×10^{-3}	0.050	2.341	0.9176	0.025

Table 3.23

Drift Loss and Pressure drop Costs for FD fan with single stage WDE

Supply Voltage = 230 V AC
Entering Air Velocity = 11.66 m/s

Inclination Angle θ (degree)	Specific Drift loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (Costs of water lost (paise)	C_2 (Costs of Pressure drop (paise)
15	1.0×10^{-3}	2.667	4.798	0.599	2.170
30	1.4×10^{-3}	2.54	6.7175	0.838	2.066
45	2.0×10^{-3}	1.18	9.596	1.187	0.826
60	2.2×10^{-3}	0.508	10.55	1.312	0.4133
90	2.6×10^{-3}	0.127	13.43	1.675	0.1033

Table 3.24

Drift Loss and Pressure drop Costs for FD fan with double stages WDE

Supply Voltage = 230 V AC
Entering Air velocity = 11.55 m/s

Inclination Angle θ (degree)	Specific Drift loss m_d (kgw/kgda)	Pressure drop ΔP (mm of H_2O)	Drift loss M_d (kg/hr)	C_1 (Costs of water lost (paise)	C_2 (Costs of pressure drop (paise)
15	0.4×10^{-3}	3.048	1.89	0.236	2.44
30	1.2×10^{-3}	2.794	5.678	0.7097	2.242
45	1.7×10^{-3}	1.27	8.045	1.005	1.019
60	1.9×10^{-3}	0.762	8.991	1.123	0.6115
90	2.4×10^{-3}	0.127	11.35	1.418	0.1019

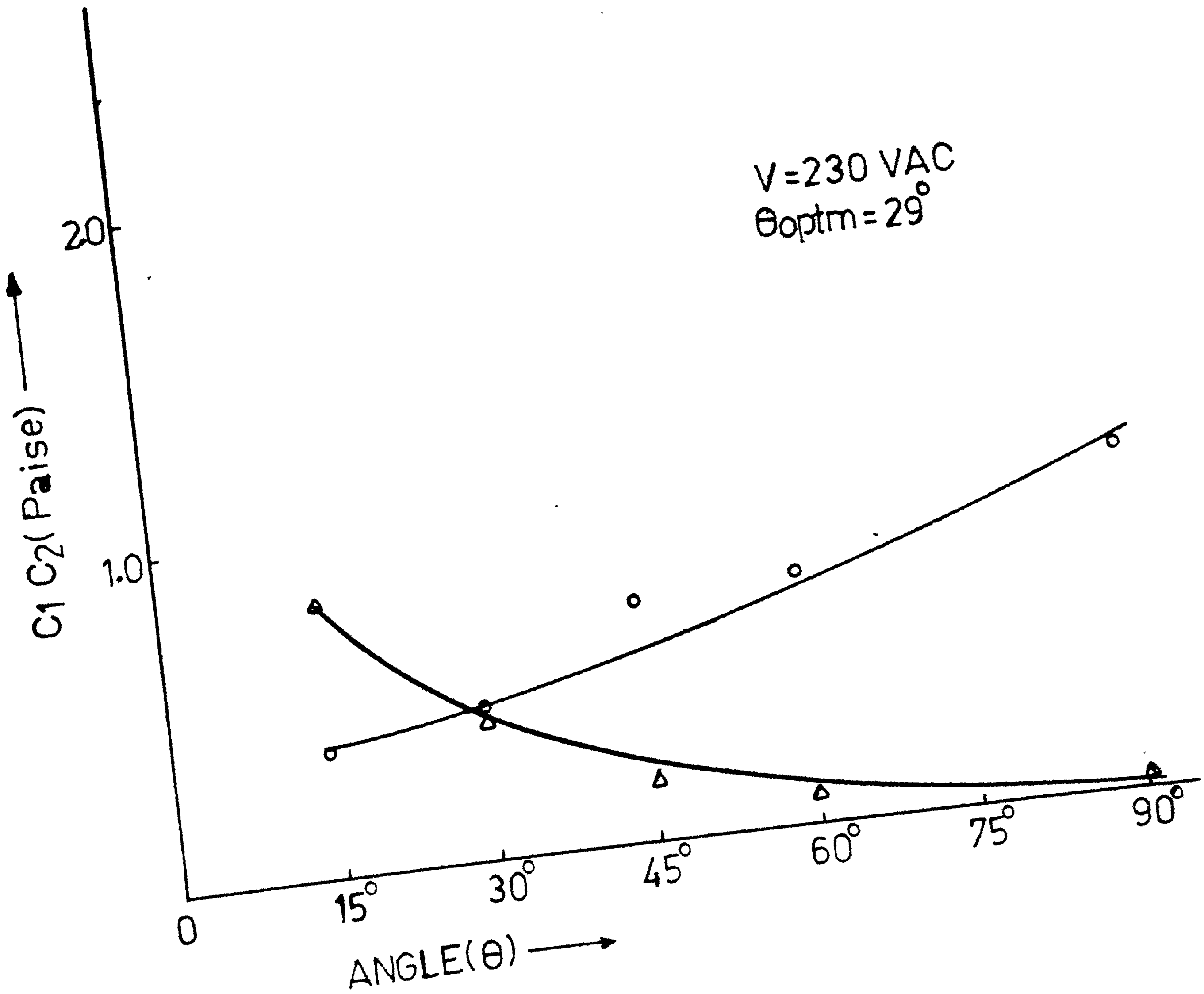

Fig. 3.23

C_1, C_2 vs Inclination angle for ID fan with single stage WDE

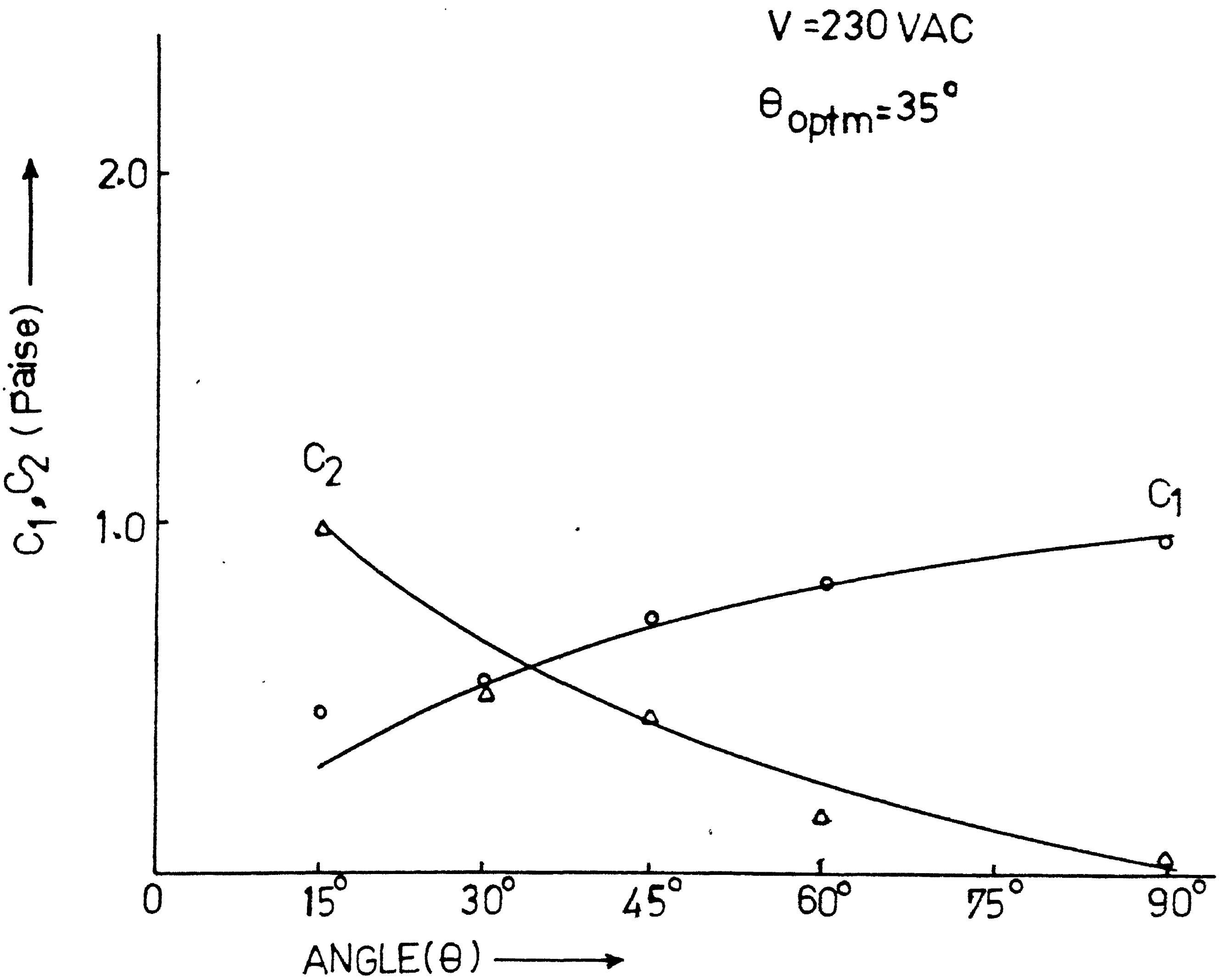

Fig. 3.24

C_1, C_2 vs Inclination angle for ID fan with double stage WDE

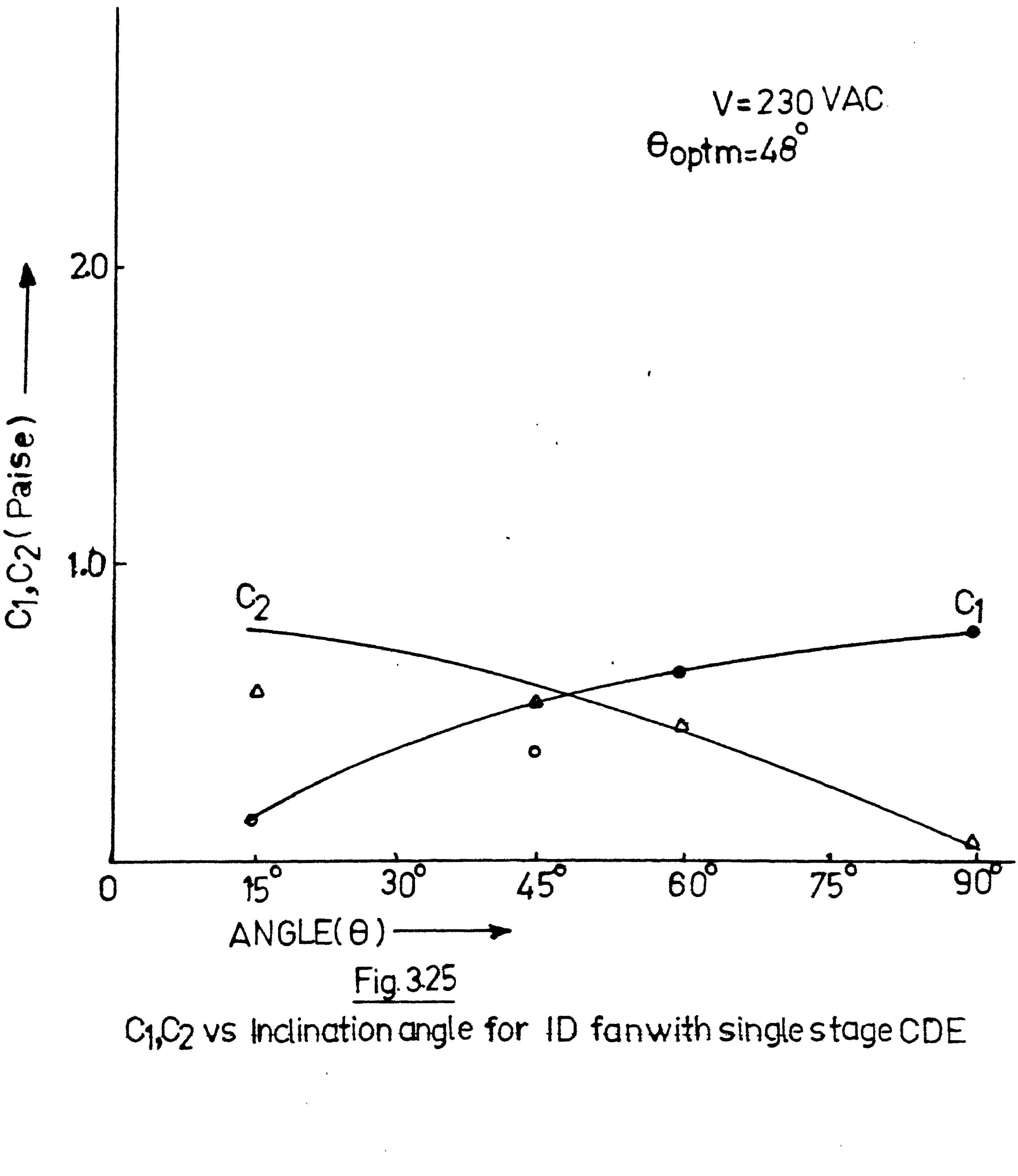

Fig. 3.25

C_1, C_2 vs Inclination angle for ID fan with single stage CDE

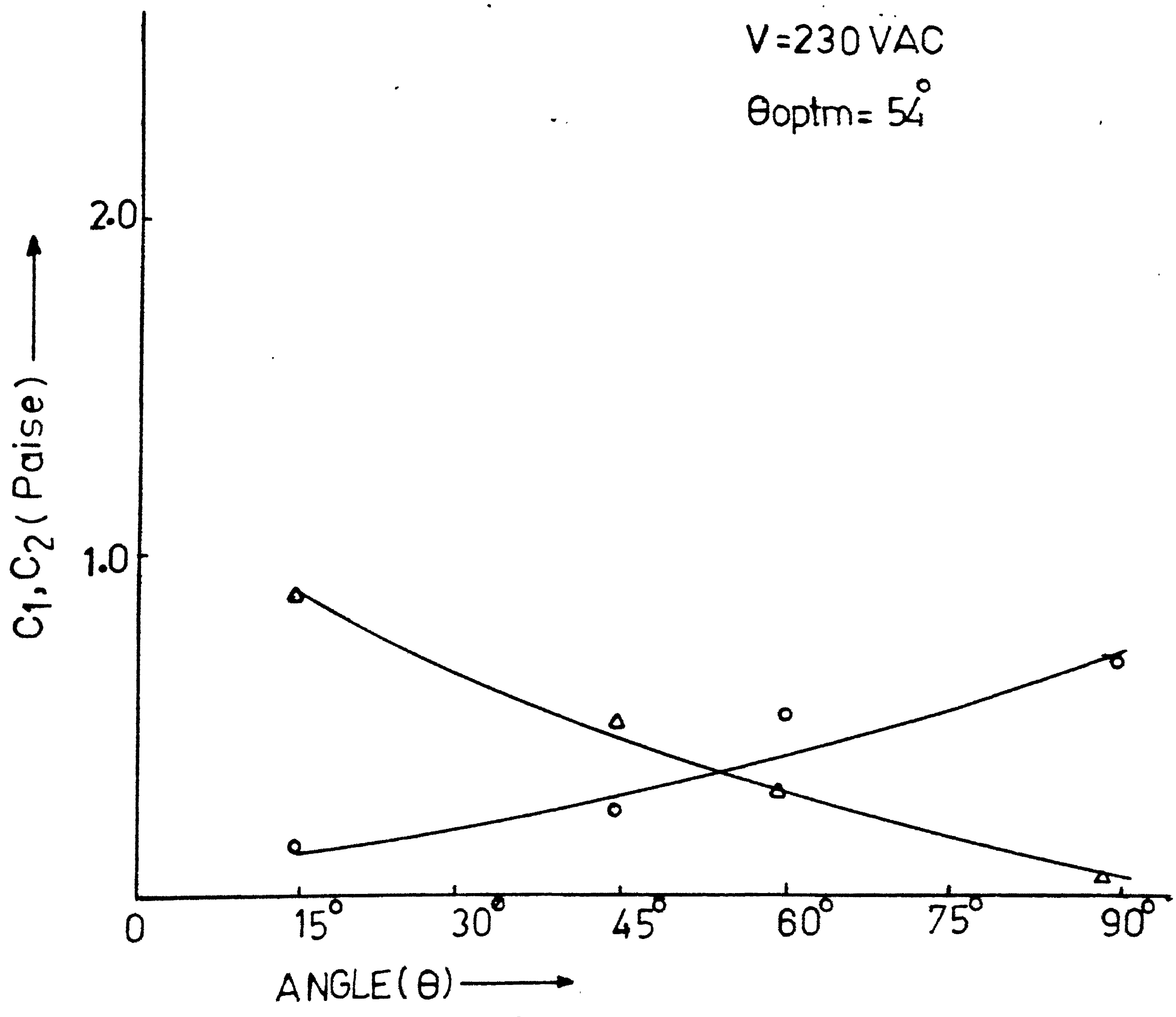

Fig. 3.26

C_1, C_2 vs Inclination angle for ID fan with double stage CDE

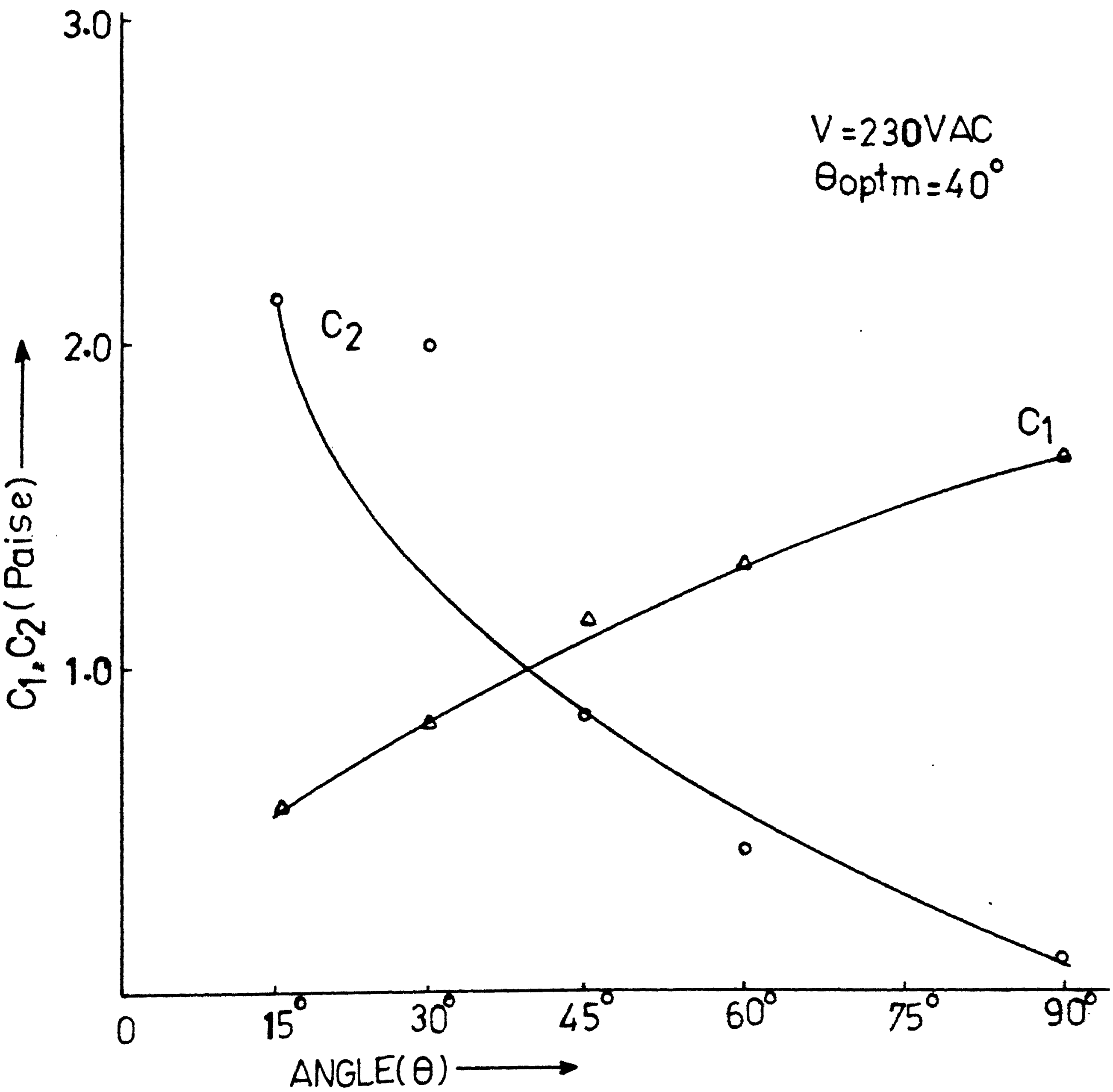

Fig. 3.27

C_1, C_2 vs Inclination angle for FD fan with single stage WDE

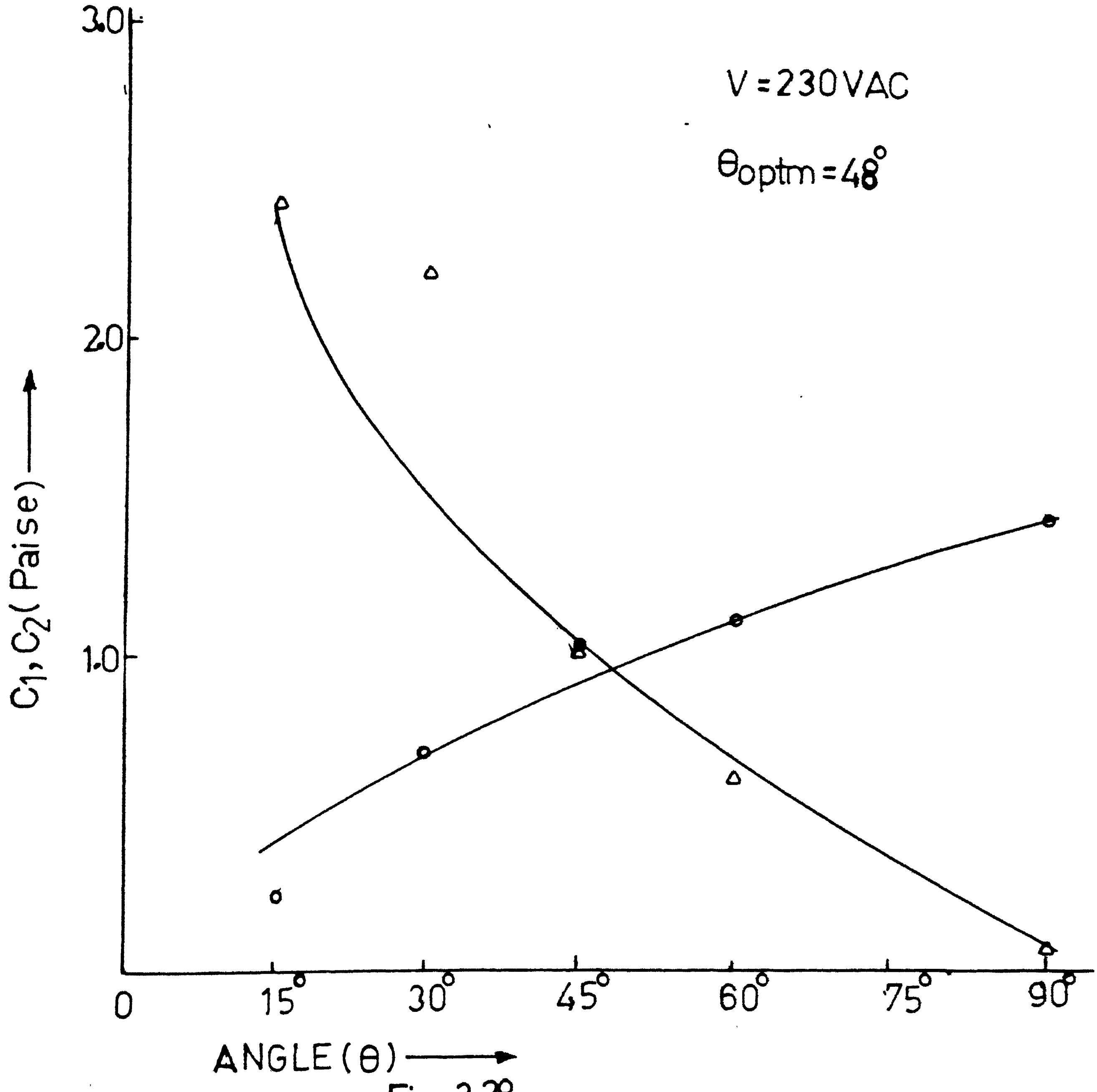

Fig. 3.28

C_1, C_2 vs Inclination angle for FD fan with double stage WDE

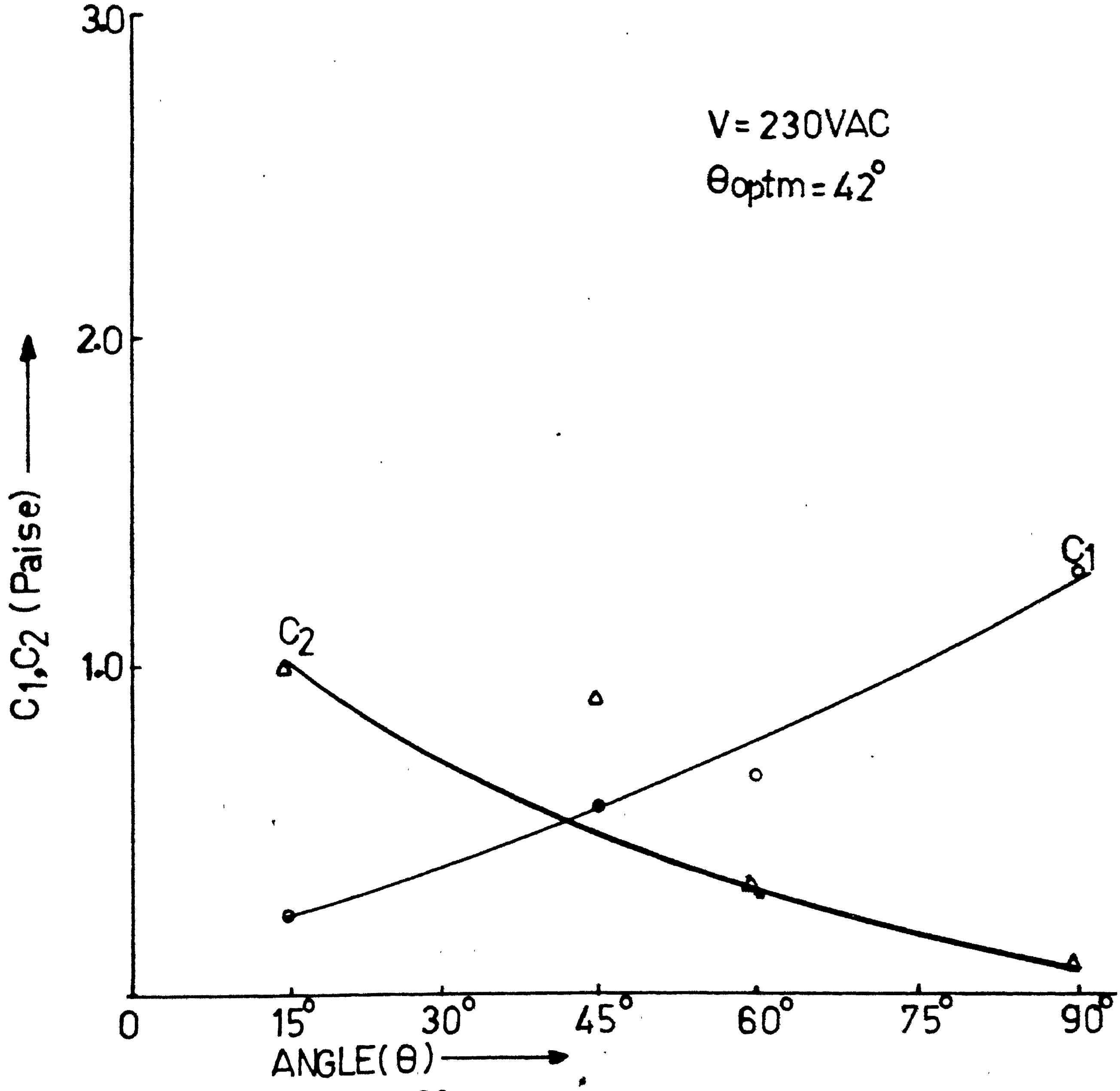

Fig. 3.29

C_1, C_2 vs Inclination angle for FD fan with single stage CDE

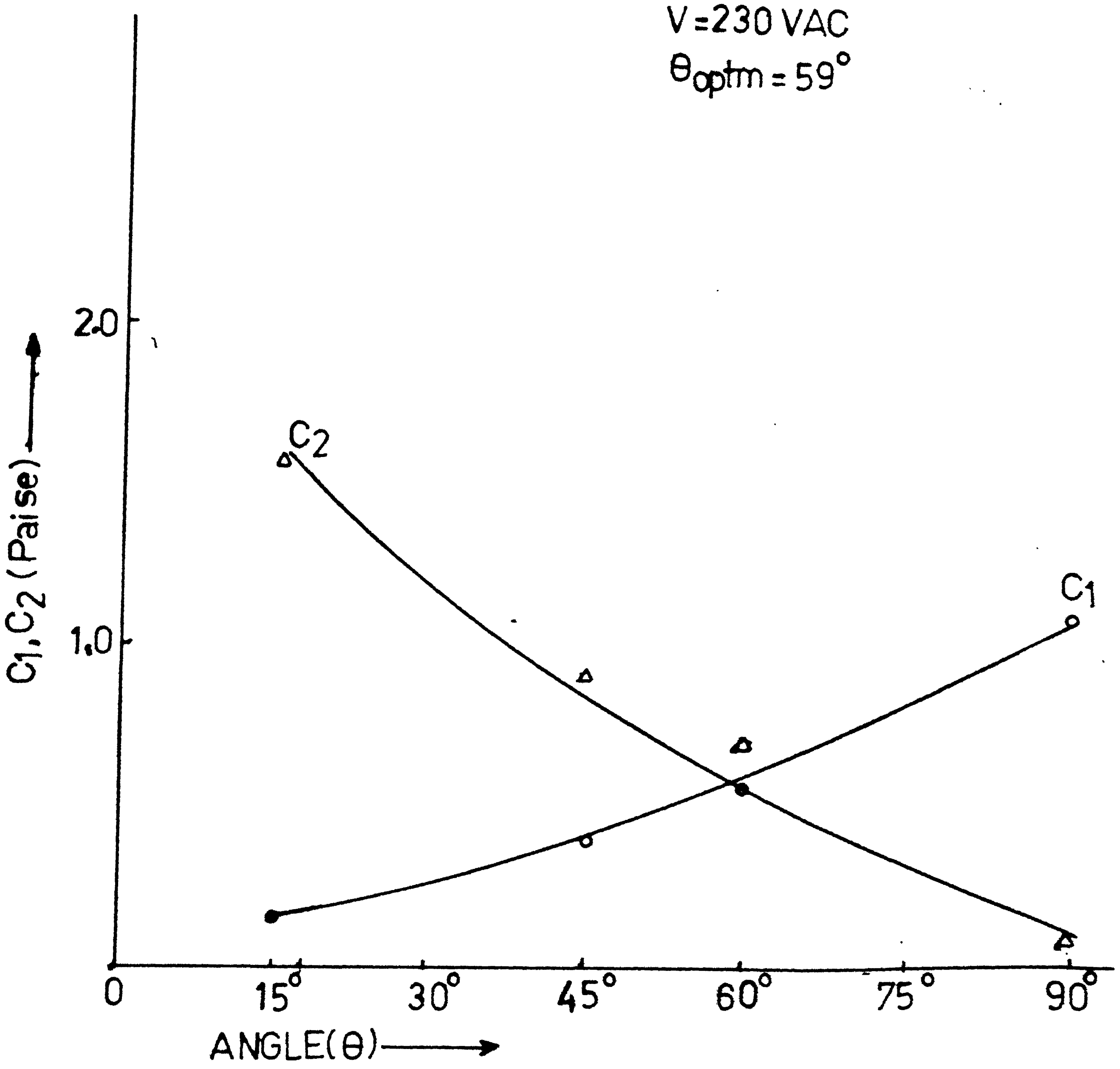

Fig. 3.30

C_1, C_2 vs Inclination angle for FD fan with double stage CDE

C_1, C_2 and $\dot{M}_d$ are given in Tables 3.18 to 3.24. The costs C_1 and C_2 are plotted versus θ (inclination angle) in Figures 3.23 to 3.30. It is found that for a particular value of θ for a given case, the values of C_1 and C_2 become equal, i.e., the cost of water lost to the atmoshphere per hour in the form of drift is same as the cost of power lost to overcome the pressure drop. This gives the optimum value of 'θ' (orientation angle) for drift eliminators for which the use of drift eliminator for a particular case becomes cost effective.

Similarly if the cost of pressure drop increases due to the increase in the capacity,cost of power or increased resistance due to algae deposits etc., the optimum value of 'θ' will be higher. It is also seen that for a given case, optimum value of 'θ' with single stage of drift eliminators will be different than that with multiple stages of drift eliminators. As the number of stages goes up, the optimum value of 'θ' also increases.

It can also be seen from these figures that for a given power tariff, if the drift increases for larger capacity of the unit, higher water circulation rate etc., the optimum θ will be lower.

3.5 CONCLUSIONS

1. Drift loss and pressure drop are more with FD fan as compared to those with ID fan.

2. Drift loss and pressure drop are less for concrete drift eliminators than those for wooden drift eliminators.

3. Orientation of drift eliminators affects the performance of evaporative condenser. A higher inclination angle to the horizontal improves the COP of the system.

4. As the number of drift eliminator stages increases, the drift loss decreases, but it leads to a higher pressure drop across the drift eliminator stages leading to an increase in the fan power input.

5. An optimum angle of orientation for drift eliminators for single stage as well as multiple stages can be found out on the basis of cost analysis. Following optimum values of 'θ' were found out in the present investigation for different cases listed below.

(I) $\theta = 29^{o}$ ID fan, single stage eliminators, supply voltage, 230 V AC, (Fig. 3.23).

(II) $\theta = 36^{o}$ ID fan, double stage wooden drift eliminators, supply voltage, 230 V AC, (Fig. 3.24).

(III) $\theta = 48^{o}$ ID fan, single stage concrete drift eliminators, supply voltage, 230 V AC, (Fig. 3.25).

(IV) $\theta = 54^{o}$ ID fan, double stage concrete drift eliminators, supply voltage 230 V AC, (Fig. 3.26).

(V) $\theta = 40^{o}$ FD fan, single stage wooden drift eliminators, supply voltage 230 V AC, (Fig. 3.27).

(VI) $\theta = 48^{o}$ FD fan, double stage wooden drift eliminators, supply voltage 230 V AC, (Fig. 3.28).

(VII) $\theta = 42^{o}$ FD fan, single stage concrete drift eliminators, supply voltage, 230 V AC, (Fig. 3.29).

(VIII) $\theta = 59^{o}$ FD fan, double stage concrete drift eliminators, supply voltage, 230 V AC, (Fig. 3.30).

In industries, normally the angle of inclination used for drift eliminators is 45° which is close to angles found with the single stages used for different cases in the present experimental work. Generally it will suffice to use only single stage of drift eliminators, but due to safety considerations double stages are employed in big units.

3.6 SUGGESTIONS FOR FUTURE WORK:

The following additional research work can be takenup on the present test rig.

1. The test rig can be used for the testing of evaporative condensers.
2. The set up can be used for conducting experiments with drift eliminators of other geometrical shapes such as conical, corrugated and perforated, and data can be used for determining corelating factors for these geometrical shapes of drift eliminators.
3. By using different types of nozzles, one can also study the effect of mode of spray on system performance.
4. The set up can be used to determine the optimum values of fan speed and water circulation rate for a given capacity of refrigeration system using evaporative condenser.

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APPENDIX

Inlet duct cross-sectional area = 0.1017 m^2.

Voltage V	Air flow rate m^3/min FD fan	Air flow rate, m^3/ min ID fan
160	45.520	37.832
180	46.751	38.625
200	52.843	40.639
220	56.931	42.714
230	61.020	45.520